V.S. Bureau of naval personnel

INTERNAL COMBUSTION ENGINES

UNITED STATES NAVAL RESERVE MIDSHIPMEN'S SCHOOL

U. S. S. "PRAIRIE STATE"

1942

Revised June 1942

Bureau of
U.S. [illegible]

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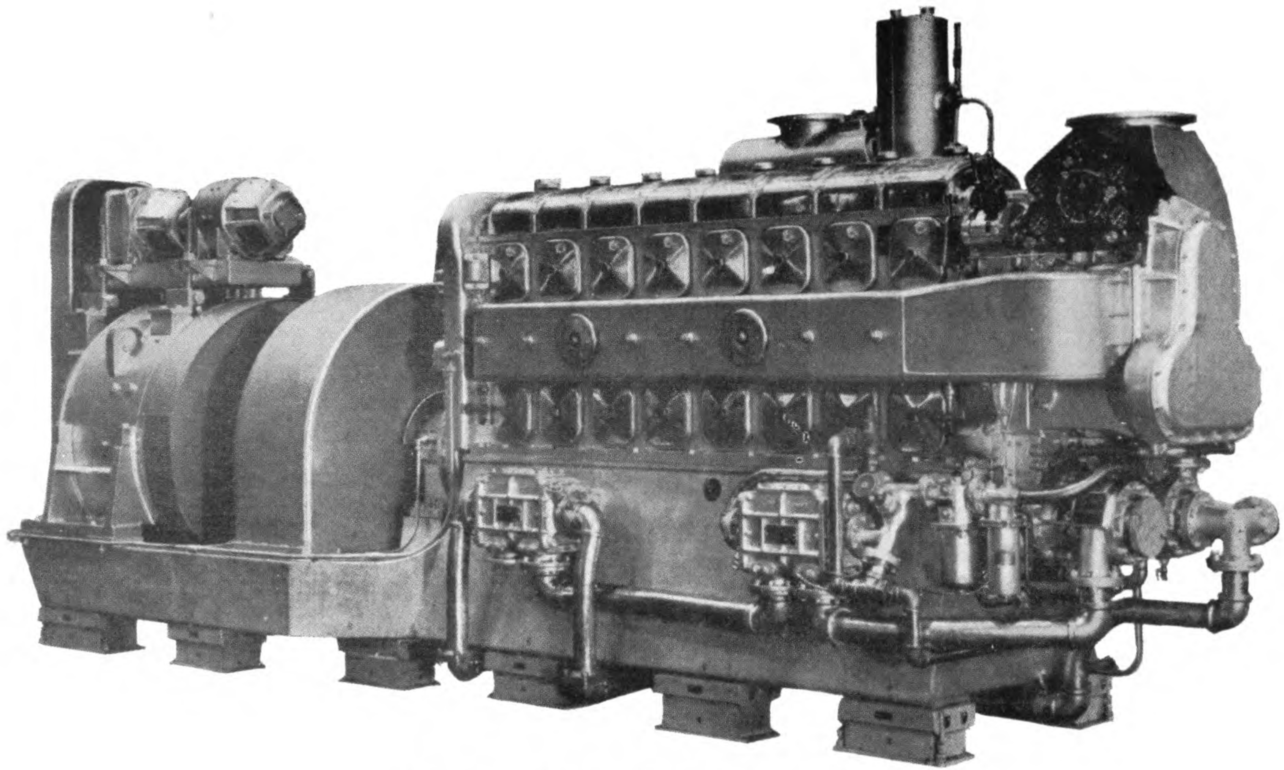

TYPICAL DIESEL-GENERATOR SET

SHOWN ABOVE IS A GENERAL MOTORS DIESEL ENGINE, DRIVING A GENERAL ELECTRIC 0 TO 540 KW GENERATOR. ENGINE DATA: TWO CYCLE, 8 CYLINDERS, 6½" BORE, 7" STROKE, SOLID INJECTION, AIR STARTING.

FOREWORD

"Internal Combustion Engines" was originally prepared by the staff of engineering instructors aboard the training ship U. S. S. *Prairie State.* Particular credit for the work is due Lt. William B. Tucker, U. S. N. R. The book has since been used in classroom work at the midshipmen's school. In June 1942 additions were made by the Bureau of Naval Personnel to adapt it for use as a text for the Diesel Engineering Correspondence Course. Much of the additional information was obtained from the Bureau of Ships, whose cooperation greatly simplified the task.

The Bureau of Naval Personnel presumes that many of the officers who take the Diesel Engineering Correspondence Course will have had little or no previous experience with internal combustion engines. Consequently, the first few lessons have been made elementary, to familiarize the student with the moving parts of an engine, and its auxiliary systems. Subsequent lessons present the fundamentals of Diesel theory, operation, and maintenance, as applied to Naval installations. Much practical information can be gained from the course by the student who applies himself.

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Chapter I. SKETCHING

All naval officers must be instructors. One of an officer's most important duties is the instruction of personnel. It is usually impossible to take machinery apart for purposes of instruction, and clear graphic portrayal therefore becomes a matter of primary importance. The art of sketching must particularly be developed by engineer officers. To this end, many examination questions will begin, "Sketch and describe . . ." Sketches intended for instruction must strive to show the principle of a piece of equipment or machinery. They need not be accurate as to scale, and exaggeration is frequently expedient.

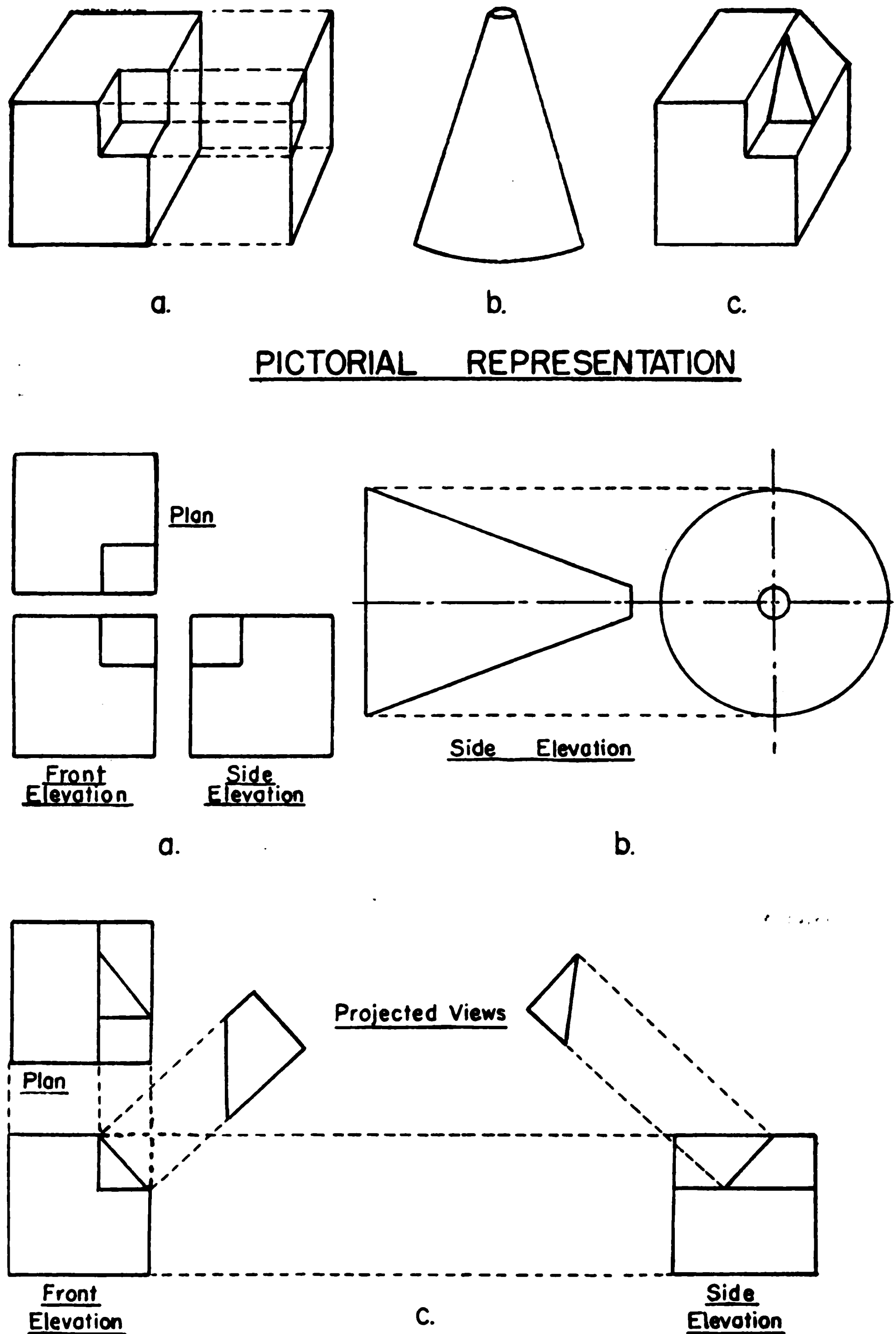
In the course of regular maintenance and in the temporary or permanent repair of battle damage, it is necessary for the ship's company to do minor mechanical construction and machine work. The engineer will be called upon to make working sketches or drawings for the ship's machinists. This will require a somewhat different type of sketching. Since the practical aspects of construction as well as the basic principles of operation must be duly provided for.

It is the purpose of this assignment to cover briefly a few of the basic principles and conventions of sketching and of mechanical drawing.

TYPES OF DRAWINGS

VIEWS

Pictorial representations are usually perspective drawings showing an object as it would normally appear to an observer viewing it from some point of vantage. In general, none of the lines on such a drawing appear in their true length. Dimensions therefore cannot be scaled from such a drawing, and it is of little value to a machinist. Resort is made, instead, to the device of projecting the outlines of parts onto convenient planes located parallel to surfaces of important parts or to their axis of rotation if they are surfaces of rotation. Figure 1 shows this method of approach for a large cube with a smaller cube cut from one corner, for a truncated cone, and for an irregular alteration of a cube. A vertical plane parallel to one side of the cube in example 1a shows the method of projection. This plane, laid flat with the paper, forms the side elevation of the plane projection. In the last of the three examples, two partial projections have been made to show the true shape of the oblique surfaces.



PLANE PROJECTIONS

Figure I.

DEVELOPMENTS

A development or a developed projection will sometimes illustrate a type of construction more clearly than a plane projection. A development of a surface generated by the rotation of a straight line about a single axis may be made by cutting the surface and laying it out as a plane figure. Developed projections are made on cylindrical surfaces by projecting the outlines of the parts radially from the axis of the cylinder, and then developing the cylinder into a plane figure. This is best shown by an example. Figure 2 shows the development of the surface of a truncated cone (such as a metalsmith would make to cut sheet metal for a funnel) and of the periphery of a ratchet wheel.

HIDDEN SURFACES

It is often impossible to choose planes for projection so the edges of all important planes are visible from the plane. The edges of such hidden surfaces are represented by dotted lines. An example of this, a cube with a hole bored in one face, is shown in figure 3.

SECTIONS

Complicated parts are frequently difficult to represent with projections onto external planes alone, and can be more easily depicted by projection onto a plane passing through solid material. The areas so cut are indicated by hatching the surface with a pattern of lines. As an example, a sectioned view of a wrist pin is shown in figure 4.

> Standard patterns may be found in several references which purport to be accepted types of hatching which identify the nature of the material cut in a section. In practice, these codes are not regularly followed, partly because simple hatching is more quickly done and is hence less expensive.
>
> The section lines of adjacent separate parts are usually hatched in opposite directions to provide contrast.
>
> The originals of mechanical drawings are rarely used directly, and the available reproduction processes are nearly all limited to high contrast monochrome prints. The use of washes or colors for shading is therefore in disfavor among mechanical draftsmen. For instruction, however, drawings which can be used as originals may often be clarified by shading with colored pencil.

RANDOM VIEWS AND SECTIONS

Random views are frequently used in special cases, notably in textbook illustrations where it is desirable to put as much information as possible into a single figure. They are obtained by cutting a machine

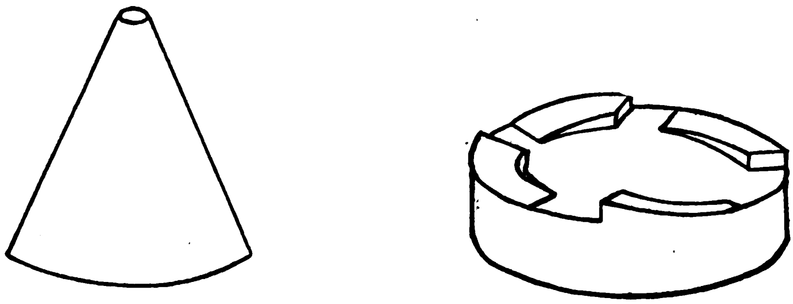
with a series of planes, some of which may be sections and others projections, so that the most interesting parts are shown in each area of the drawing. Such random views should be accompanied by another view in a plane at right angles with the random planes and showing their relation to reference planes in the machine, but in practice the key drawing is usually omitted. A random section through an internal combustion engine piston is shown in figure 5. The piston, in this case, is cut by a plane passing through the axis of the wrist pin bearing on the left-hand side of the figure, and normal to this plane on the right-hand side of the figure. Neither half of the two halves of the figure presents a true section. The webs in the crown of the piston are actually cut by the section plane, but in order to show the thickness of the crown and of the other webs, the section is passed just in front of the centerline webs in their own locality.

STANDARD CONVENTIONS

Certain common machine elements are rather difficult to depict accurately, and except for special illustrative drawings, they are represented by standard conventions. Gears may be represented as in figure 6. The outer circle represents the tip circle, the dot-and-dash line the pitch circle (the diameter of a smooth cylinder which could replace the gear in the gear train), and the dotted line the base circle. In some cases, one or two gear teeth are drawn to show the profile. For diagrammatic purposes, gears are often shown simply with an interrupted line representing the pitch circle.

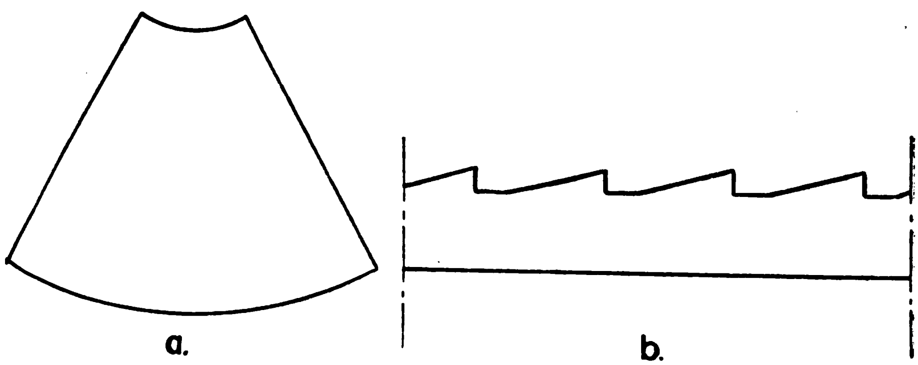
Springs are most often depicted by passing a plane through the axis of the spring, sectioning the wires of the spring at right angles to their axes. The wires behind the section plane are projected onto it. An example is shown in figure 7a. It is not necessary to draw in all of the turns of the spring wire. Frequently only the end turns are drawn and the presence of the missing turns is indicated by a centerline passed through the centers of the sectioned wires. This is illustrated in figure 7b. For diagrammatic purposes, springs may be shown as a simple zigzag, as in figure 7c, or as a row of dots, figure 7d.

Screws are especially difficult to draw accurately, and the conventions shown in figures 8a and 8b are nearly always used. The convention in figure 8b is a form of random section, the plane cutting the female part on the axis of the hole, but passing in front of the male part. This differentiates between male and female threads, and between left-hand and right-hand screws.



a. b.

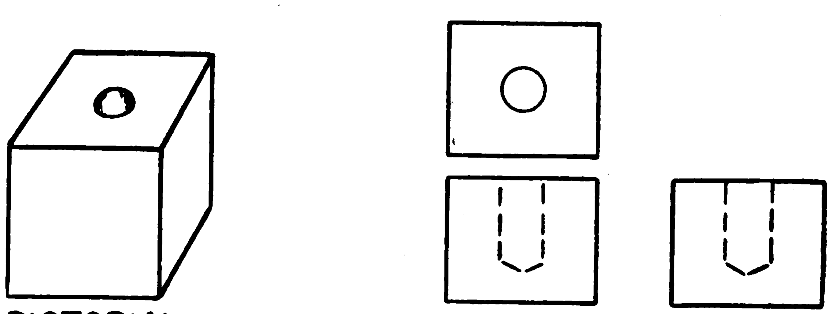
PICTORIAL REPRESENTATION



a. b.

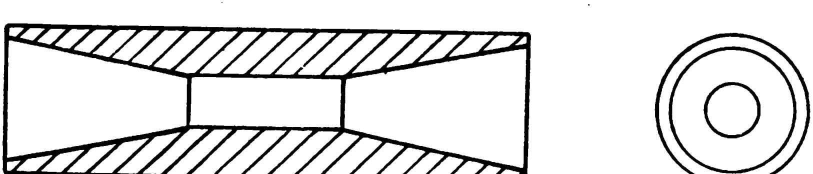
DEVELOPMENTS

Figure 2.



PICTORIAL PLANE PROJECTIONS

Figure 3.



LONGITUDINAL SECTION END VIEW

Figure 4

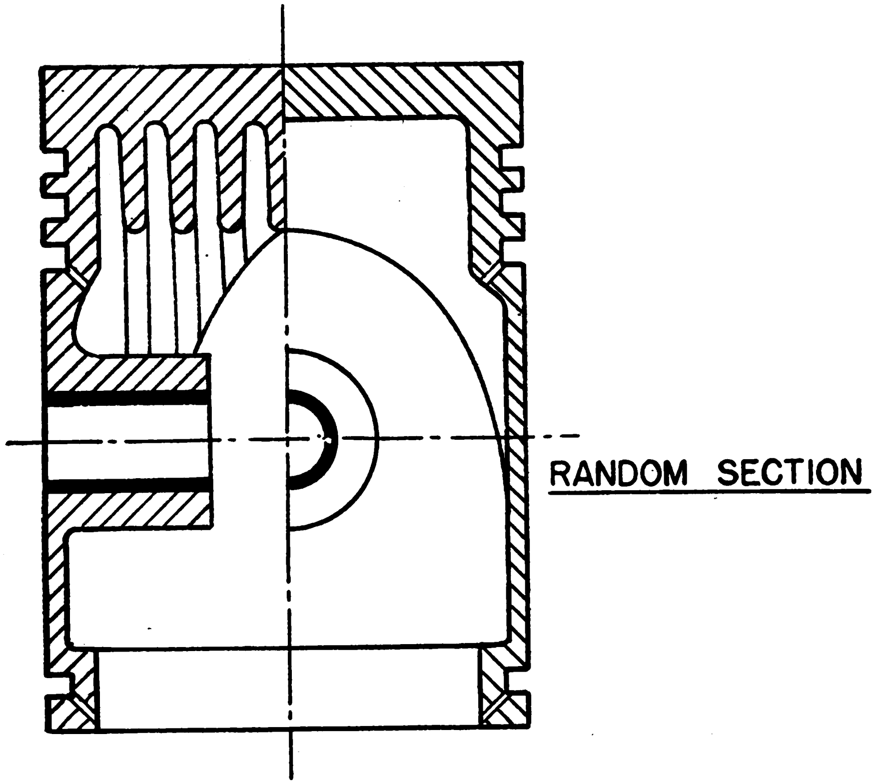
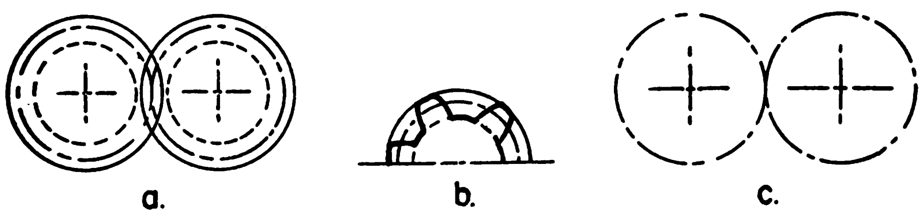
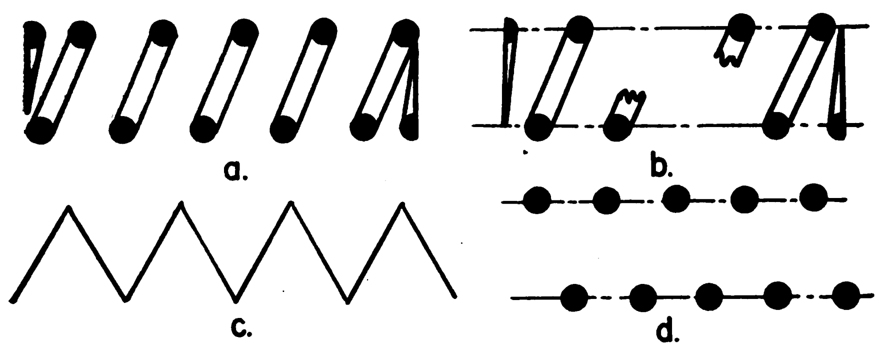


Figure 5.



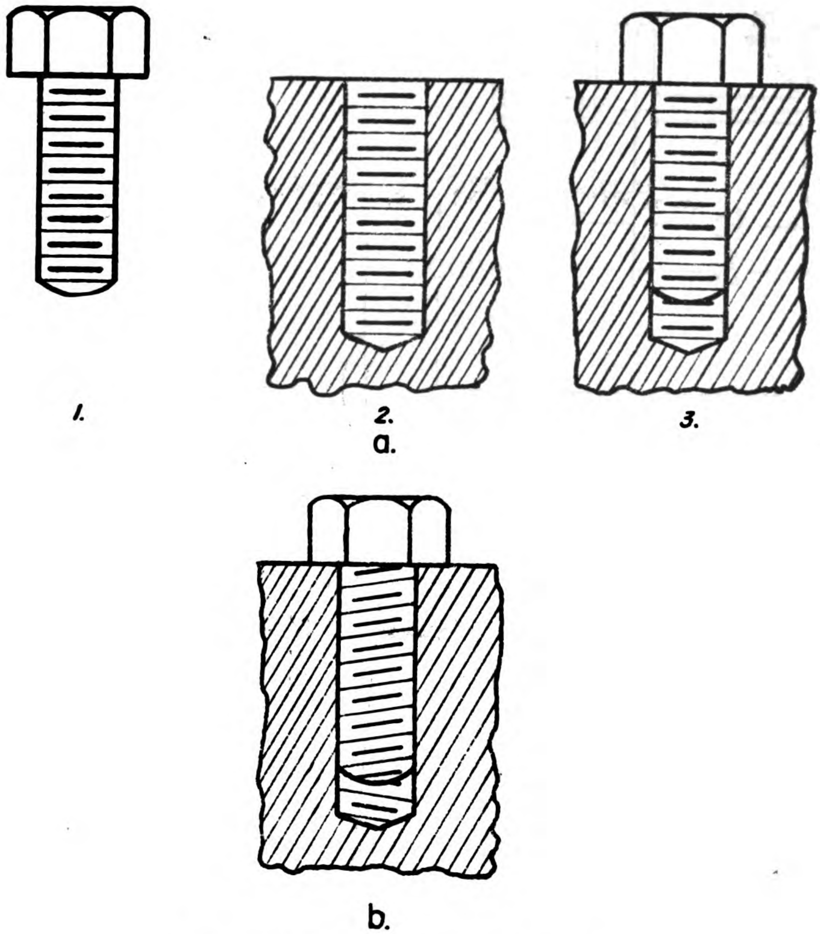
GEAR CONVENTIONS

Figure 6.



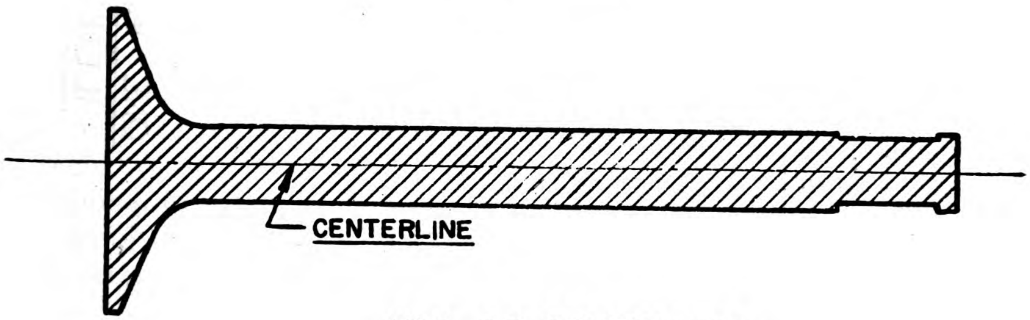
SPRING CONVENTIONS

Figure 7.



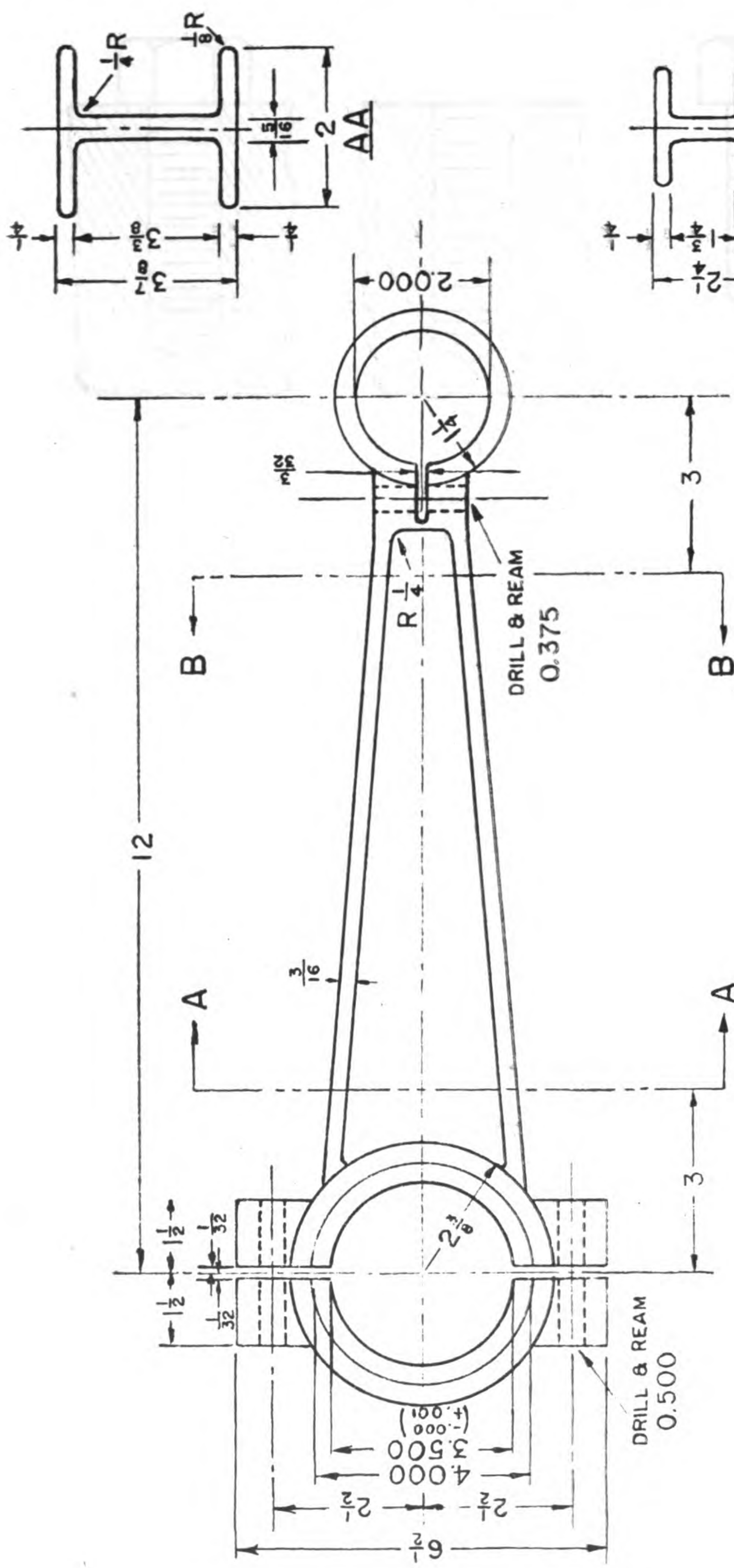
THREAD CONVENTIONS

Figure 8.



USE OF CENTERLINE

Figure 9.



ONE VIEW OF A CONNECTING ROD ILLUSTRATING DIMENSIONING.

Figure 10.

CENTERLINES

Drawings are constructed from some base or reference line, and in cases where a part or an element is symmetrical, it is common to use a centerline for a base. This is especially true of surfaces of revolution. By convention, a dot-and-dash line is used to indicate a centerline. As an example, a section through a valve is shown in figure 9.

DIMENSIONS AND TOLERANCES

The proper dimensioning of a drawing is an art that requires much experience. The conventions used and the handling of some specific problems are shown in the drawing of a connecting rod, figure 10. The general rules are to dimension the part from some surface which can be finished machined as the initial step in making the part. Avoid as much as possible making it necessary for the machinist to add a number of odd fractional dimensions to locate some element. Dimension symmetrical parts from centerlines, particularly those which are bounded by surfaces of revolution, such as parts to be made on a lathe, holes to be bored, etc.

Dimensions must be accompanied by tolerances, either explicit or implied. The tolerance is the amount which an acceptable part may deviate from the design dimensions. Working to exacting tolerances is slow and expensive, so it is important to use as large a tolerance as is consistent with a shipshape job and with the purpose to which the part will be put. Interchangeable parts must always be made to a close tolerance, even though in the individual case greater deviations could be accepted.

Relatively unimportant dimensions are written as fractions, and a tolerance of ± 0.005 to ± 0.010 inch is expected on small parts. More critical dimensions are written in decimals, and a tolerance of ± 0.001 for dimensions of 6 inches or less is reasonable. When greater precision is required, the dimension is given in decimals and the tolerance is specified. Comparatively few machines or machinists are capable of producing single parts to a tolerance of less than 0.0002 inch. Working in air-conditioned rooms with precision tools, machining, grinding, and lapping, it is possible to finish some parts to tolerances measured as millionths of an inch. The pistons of Diesel injection pumps, which develop pressures up to 10,000 pounds per square inch, are so machined and operate in their cylinders without packing of any sort.

STANDARD PARTS

Many common parts have been standardized, and deviations from these standards should not be made except in very unusual cases.

Among such parts are screws, bolts, and nuts, dovetail dimensions, bearings and bushings, gears, pipe couplings and fittings, and many others.

There are two series of thread standards—National Fine (NF) and National Coarse (NC). NF machine screw threads are used where vibration is severe. NC threads are used for common construction.

REFERENCE

One of the best references on machine construction and practical machine work is Machinery's Handbook, published by The Industrial Press, New York, N. Y. Not only detailed information on the subject of this assignment, but a great deal of practical information relative to the repair and adjustment of machine elements of all kinds is contained in this handbook.

Chapter II. SCREW FASTENINGS

FORMS OF SCREW THREADS

For screw fastenings the *American* (*National*) *Standard* V-thread is used in this country. This thread was formerly called the United States Standard or the Seller's Standard, and these names are still encountered. This form of thread (see fig. 1) employs a 60° V which

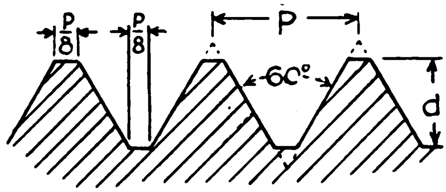


Fig 1 American Standard Thread Form Profile

is flattened at the top and bottom for additional strength. The pitch, p, is the axial distance from a point on one thread to the corresponding point on the next thread. For single thread screws, the pitch is the distance that the nut advances along the axis of the screw for each revolution of the nut. Also the pitch equals the reciprocal of the number of threads per inch of the screw. The depth of the thread, d, is $0.6495p$. The flats are equal to ⅛ of the pitch ($p/8$).

Two thread series are used, a coarse-thread series and a fine-thread series with basic dimensions and threads per inch as shown in table I. The coarse-thread series is used for general engineering work where rapid and easy assembly are desired. All studs seating in cast iron, aluminum or other comparatively weak material have coarse threads to give the threads in the tapped hole additional strength. The fine-thread series is used for automotive and airplane work.

In addition to the American Coarse and Fine series of threads a third series, the S. A. E. Extra-Fine, is used where light sections, fine adjustments, and vibration are important factors.

TYPES OF BOLTS, NUTS, SCREWS, ETC.

Bolts, cap screws, machine screws, set screws, and studs are examples of screw fastenings commonly used for joining machine parts.

Machine bolts have a cylindrical body, the head of which is either upset forged or is formed by cutting the bolt from hexagonal or

square stock while threads are cut on the other end which is provided with a nut. See figure 2.

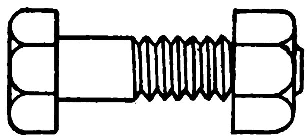


FIG. 2. American Standard hexagonal head machine bolt and nut.

Cap screws, unlike bolts, do not require a nut but screw into a tapped hole in one of the parts through a clear hole in the other, holding the parts together by the pressure exerted by the head of the cap screw on the second part. Cap screws are made with square, hexagonal, round, filister, Allen, button, flat, or oval heads. These various forms of heads are shown in figure 3.

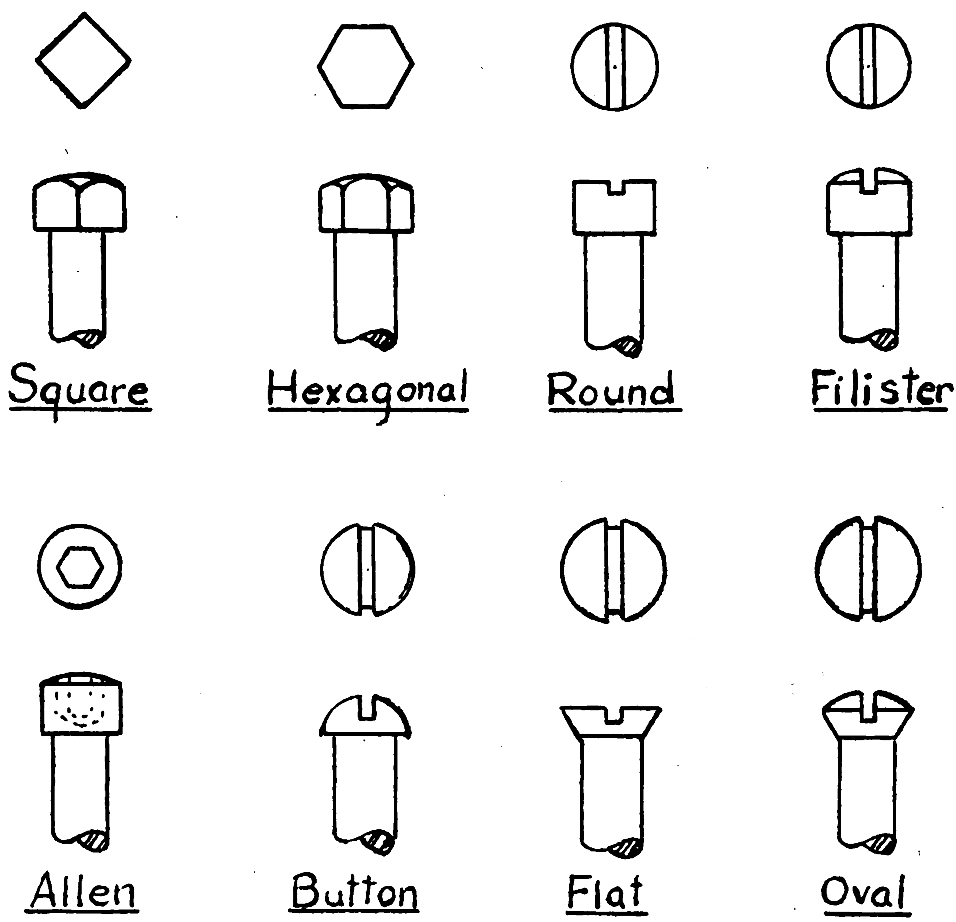


FIGURE 3. Forms of heads used on cap screws.

TABLE I

Bolt or screw diameter, inches	American Coarse (U. S. Standard, S. A. E. Coarse)				American Fine (S. A. E. Regular)				S. A. E. Extra Fine			
	Threads per inch	Root diameter, inches	Tap drill size		Threads per inch	Root diameter, inches	Tap drill size		Threads per inch	Root diameter, inches	Tap drill size	
			Decimal	Nominal			Decimal	Nominal			Decimal	Nominal
1/4	20	0.1850	0.1850	No. 13	28	0.2036	0.2040	No. 6	36	0.2139	0.2130	No. 3.
5/16	18	.2403	.2402	6.1 mm	24	.2584	.2598	6.6 mm	32	.2719	.2720	I.
3/8	16	.2938	.2950	M	24	.3209	.3228	8.2 mm	32	.3344	.3346	8.5 mm.
7/16	14	.3447	.3464	8.8 mm	20	.3725	.3740	9.5 mm	28	.3911	.3903	25/64.
1/2	13	.4001	.4040	Y	20	.4350	.4375	7/16	28	.4536	.4531	29/64.
9/16	12	.4542	.4531	29/64	18	.4903	.4921	12.5 mm	24	.5084	.5118	13.0 mm.
5/8	11	.5069	.5118	13.0 mm	18	.5528	.5512	14.0 mm	24	.5709	.5709	14.5 mm.
3/4	10	.6201	.6250	5/8	16	.6688	.6693	17.0 mm	20	.6850	.6875	11/16.
7/8	9	.7307	.7344	47/64	14	.7822	.7812	25/32	20	.8100	.8125	13/16.
1	8	.8376	.8437	27/32	14	.9072	.9063	29/32	20	.9350	.9375	15/16.
1 1/8	7	.9394	.9375	15/16	12	1.0167	1.0156	1 1/64	18	1.0528	1.0469	1 3/64.
1 1/4	7	1.0644	1.0630	27 mm	12	1.1477	1.1417	29.0	18	1.1778	1.1811	30.0 mm.
1 1/2	6	1.2835	1.2812	1 9/32	12	1.3917	1.3906	1 25/64	18	1.4278	1.4375	1 7/16.
1 3/4	5	1.4902	1.4961	38 mm					16	1.6778	1.6718	1 43/64.
2	4 1/2	1.7113	1.7126	43.5 mm					16	1.9278	1.9218	1 59/64.

Machine screws are, strictly speaking, cap screws that are provided with a slotted head for a screw driver. They are used for the same purpose as bolts and cap screws in small work, particularly sheet metal. They are identified by gage number, ranging from No. 0 (0.06″ diameter and 80 threads per inch) to No. 30 (0.45″ diameter).

Set screws are used for holding two parts in relative position, being screwed through one part and having the point set against the other. They are made with square heads or made headless to eliminate the hazard of sharp projections on moving parts. The principal distinguishing feature of set screws is the form of the point. Several types of points are shown in figure 4. The points are generally hardened. Set screws used as fastenings should be used only on lighter loads.

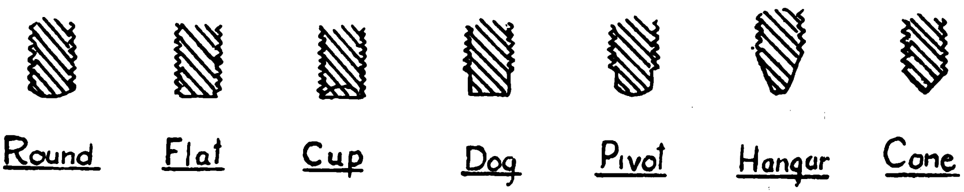


FIG. 4. Types of set screw points.

Studs.—The stud or stud bolt, which is threaded on both ends, is used when through bolts are not suitable, for parts which must be removed frequently, such as cylinder heads, covers, etc. One end is screwed permanently into a tapped hole and the projecting studs guide the removable piece to position. With this construction the wear and crumbling of the threads in a weak material such as cast iron or aluminum are avoided.

TYPES OF NUT LOCKS

Since nuts must have a small clearance in order to allow them to turn freely, they have a tendency to unscrew. This tendency is especially evident in the case of nuts subjected to vibration. In order to prevent unscrewing, a number of different devices have been originated, some of which are listed below.

Lock nuts or jam nuts are one of the cheapest and most common nut-locking devices. Two nuts are used, the lower nut being screwed down tight against the piece to be secured; then the upper nut is screwed against the lower nut as tightly as the size of the bolt or stud will permit, thus developing a pressure between the two nuts which at the same time produces a tensile stress in that part of the stud (or bolt) that comes within the action of the two nuts. If one of the two nuts is one-half standard thickness, as is often the case, it may be properly placed above or below depending on the magnitude of the

pressure between the nuts compared with that between the lower nut and the piece. A half-thickness lock nut is shown in figure 5.

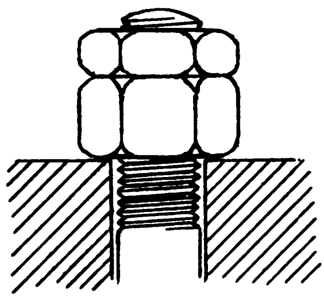


Fig. 5. Lock Nut

Cotter pins used with castellated nuts is another common and effective way of locking nuts.

Due to the necessity of turning the nut through 60° between successive locking positions, it may be impossible to obtain a tight and rigid connection without causing a high stress in the bolt or stud. Such a cotter pin should be so chosen as to fit snugly into the drilled hole in the bolt or stud and in the castellation of the nut.

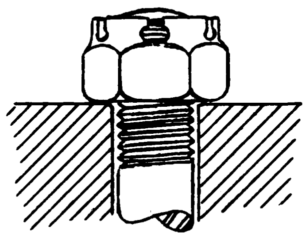


Fig. 6 Castellated nut with a cotter pin.

Lockwashers or spring washers are also common devices for locking nuts. Figure 7 shows the plain form which consists essentially of one complete turn of a helical spring placed between the nut and

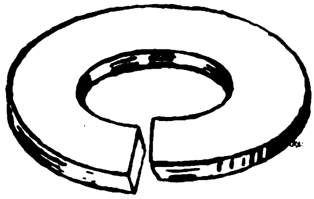


Fig. 7

Lock washer

the piece to be fastened. The nut is screwed down tightly, the washer is flattened out and its elasticity produces a pressure upon the nut, thereby preventing backing off. In addition to the plain lock washer there are numerous patented variations using the same principle.

Wire of the ordinary black variety is sometimes used as a locking device by threading it through drilled holes in the heads of cap screws. The wire must be pulled up tightly and must be so threaded as to oppose any loosening of the cap screw. Or, one cap screw can be wired to some fixed object.

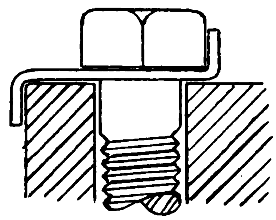


FIG. 8. Ear washer locking a capscrew.

Ear washers which may take a variety of forms are another type of nut or capscrew locking device. Figure 8 shows one form of ear washer locking a capscrew. With this device the capscrew (or nut) is screwed up tightly on the flat ear washer, one ear of which has been turned down on the fixed part to hold the washer itself. Then another ear is turned up against a flat of the capscrew head (or nut) thus locking the nut to the washer which in turn has been locked to the part. Care must be exercised in bending the washer ears so that they are not too greatly work-hardened or gouged, and yet are firmly seated against the nut flat or part. Some ear washers fit over two capscrews or studs and are thus restrained. Other ear washers fit over splines or have ears fitting into drilled holes for positioning the washer. In all cases one or more ears are turned up against the nut or capscrew.

Many other nut locking devices such as collar nuts, spring wire locks, split nuts, etc., are also used.

FLAT WASHERS

The function of a flat washer is to provide suitable bearing surface for a nut or bolt head. Washers should not be used unless the hole through which the bolt passes is very much oversize, or the nature of the material against which the nut or bolt head bears necessitates their use. When the material against which the nut or bolt head bears is relatively soft, the bearing pressure due to the load carried by the bolt should be distributed over a considerable area.

Flat washers are specified by the so-called nominal diameter, by which is meant the diameter of the bolt with which the washer is to be used.

STRESSES IN SCREWS DUE TO SCREWING UP

The stresses induced in bolts, capscrews, and studs by screwing them up tightly are a tensile stress due to the elongation of the threaded member and a torsional stress due to the frictional resistance of the threads. The effect of these combined stresses can be approximated by the relationship.

$$\text{Max. tensile stress} = \frac{S_t}{2} + \sqrt{S_s^2 + \frac{S_t^2}{4}}$$

$$\text{Max. shear stress} = \sqrt{S_s^2 + \frac{S_t^2}{4}}$$

S_t = the direct tensile stress
S_s = the direct shearing or torsional stress

However, for small screws (¾-inch in diameter or less) the stresses depend so much on the judgment of the mechanic that stress calculations are of little value.

Great care must be exercised in tightening up on small bolts, capscrews, and studs!

Chapter III. THERMODYNAMICS

1. ENERGY

All prime movers used by the Navy are heat engines and derive their power from chemical energy stored in fuels by a process involving the combustion (rapid oxidation) of the fuel. It is therefore well for us to start with a brief introduction to thermodynamics—the study of heat power engineering.

Energy is defined as the capacity for doing work. For convenience we recognize several forms in which energy may be stored.

(*a*) *Chemical energy*.—Energy stored within a material, possibly in the form of interatomic forces. The unit of chemical energy is the British thermal unit (1/180 of the heat required to heat 1 pound of of water from 32° to 212° F.).

> In general, this energy is potentially available because the oxides of the elements of the material are more stable than the material itself, and the material may therefore be oxidized (or burned) with the liberation of heat. All fuels contain energy stored as chemical energy, and all portable power plants derive their energy from this source. In the Navy, petroleum products are now used as fuels to the exclusion of all others.

(*b*) *Electrical energy*.—Energy stored within a material as the result of the extraordinary position of certain free electrons.

> This is the method of storage in a dielectric field. Chemical energy, rather than electrical energy, is stored in a battery, the method of storage being such that electricity may be produced as the result of the release of this energy. Electrical energy is not used afloat as a primary source of power, but only as an intermediate between initial derivation from storage as chemical energy and ultimate utilization as mechanical energy.

(*c*) *Potential energy*.—Energy of position stored in a system of tangible bodies as the result of their relative position and of forces acting between them. The unit of potential energy is the foot-pound, the amount of energy required to move 1 foot against a resisting force of 1 pound.

Pressure is a form of potential energy. For use in the energy equation it is evaluated as a head of fluid in feet, determined by divid-

ing the pressure in pounds per square foot by the density in pounds per cubic foot.

For our purposes we consider a tangible body as one of finite dimensions, i. e., greatly larger than a single molecule. A system is an arrangement of two or more tangible bodies, considered together with the forces acting between them.

A body weighing 250 pounds and suspended 80 feet above the earth has a potential energy of 20,000 foot-pounds relative to the earth. This stored energy may be obtained by lowering the body with suitable tackle.

Potential energy is relative.—With respect to a platform erected 70 feet above the earth, the body in the above example would have a potential energy of only 2,500 foot-pounds. It is therefore always essential to state the position of the datum (or to define the system) in order to state the amount of potential energy available in a system.

Potential energy may always be calculated as the product of force times distance.

(*d*) *Kinetic energy.*—Energy stored in a system as the result of relative motion of two or more tangible bodies. The unit of kinetic energy is the foot-pound.

A rifle bullet flying toward a wooden block has stored kinetic energy with respect to the block. If it is allowed to strike the block, this energy will be released and expended in damaging the block and the bullet.

Kinetic energy is relative. Two bullets traveling in the same direction at the same speed cannot act upon one another, and hence have no relative kinetic energy, even though each may have a large amount of kinetic energy relative to some third body which may itself be moving or stationary.

From physics we learn that for a body weighing 1 pound and moving with a velocity of U feet per second relative to a second body

$$\text{Kinetic energy} = \frac{U^2}{2g} = \frac{U^2}{64.4} \text{ foot-pounds.}$$

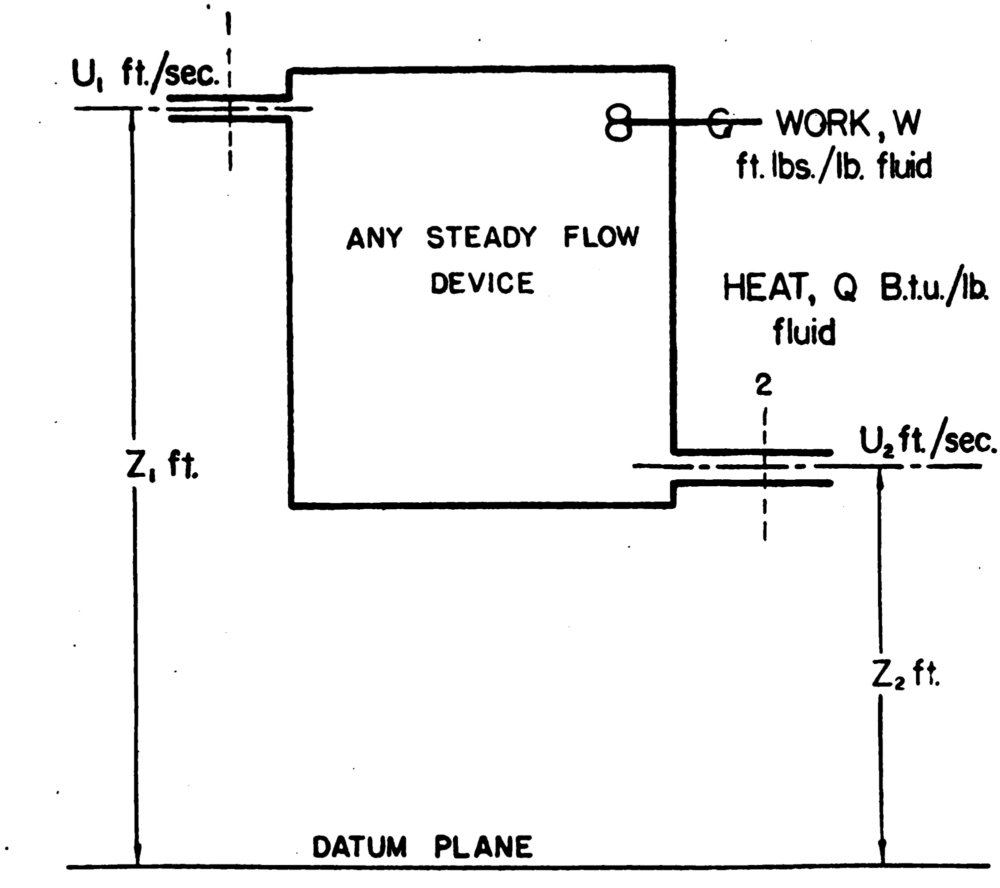
(where g is the gravitational constant in English units—32.2 feet/sec.^{2})

Kinetic energy and potential energy of position are among the highest forms of energy in that they are entirely interconvertible and may be completely converted to any other form of energy by conceivable machinery. We recognize this by frequently treating the two together.

(*e*) *Mechanical energy*.—The sum of the kinetic and the potential energy of a system. The unit of mechanical energy is the foot-pound.

(*f*) *Internal energy*.—Energy stored within a tangible body as molecular energy, and manifested by the temperature of the body. The unit of internal energy is the B. t. u.

In contrast to mechanical energy, internal energy is stored within a single body. A system is not required. Like the other forms of energy defined above, internal energy is relative, and is evaluated relative to the internal energy of the body taken as zero at some arbitrary datum temperature (usually at 32° F.).



STEADY FLOW

Figure 1.

So far we have been dealing with energy in storage. Ignoring such possibilities as the flow of electric current, we recognize two major forms of energy in transition.

(*a*) *Work*.—Mechanical energy in transition. The unit of work is the food-pound.

> A body falling in such a manner that the potential energy is not wasted but is forced to act on another body in the system is said to do work on the other body. Work, in a sense, is not tangible. Energy is said to be transmitted from an engine to a propeller along the shaft as work, but this is evident only as the result of the work that is done. Work must always be measured in terms of the effect produced.

(*b*) *Heat*.—Energy moving from one body of a system to another as the result of a temperature difference. The unit of heat is the B. t. u. (British thermal unit).

> Thermodynamics is an unusual if not a unique science in that it is based on certain laws which state that, since during long experience certain things have never been observed to happen, it follows that they never can happen. It is thus, in fact, a negative science in that it tells what cannot happen rather than defining directly what will happen.

POWER AND TORQUE

Power is the *rate* of doing work. It is the custom of engineers to use the *horsepower* as the *unit* of the rate of doing work. The numerical value of the horsepower is 550 foot-pounds per second, or 33,000 foot-pounds per minute. One horsepower-hour, or 60 x 33,000= 1,980,000 foot-pounds, is equivalent to 2,545 B. t. u.

In electrical energy, the product of 1 volt and 1 ampere is equal to 1 watt; 1,000 watts is equivalent to 1 kilowatt (kw.). *One horsepower* is equivalent to *746 watts.*

Torque is the turning moment exerted by a rotating force, and is the product of the force multiplied by its moment arm (the perpendicular distance between center of rotation and line of action of the force) measured in foot-pounds. If an engine is running at N revolutions per minute, and the engine torque is T foot-pounds, the power output of the engine will be:

$$H.P.=\frac{2\Pi NT}{33{,}000}$$

It will be seen later that the power output of an engine can also be expressed:

$$H.P.=\frac{PLAN}{33{,}000}$$

The student should remember that the N in the torque equation is the actual r. p. m. of the engine, but the N in the second equation is the actual *number* of *power strokes* per *minute* in *all cylinders.*

The first law of thermodynamics states that energy can neither be created nor destroyed, or stated in a form more convenient to our present purpose: "Heat and mechanical energy are interconvertable, but neither can be either created or destroyed."

It is possible to convert mechanical energy to heat completely, and by delicate physical experiments it has been found that *for every 778 foot-pounds of mechanical energy so converted, one B. t. u. of heat will be obtained.* Because of fundamental limitations, it is not usually possible to convert heat completely to work, but for every B. t. u. that is converted, 778 foot-pounds will be realized. This important constant is known as the *mechanical equivalent of heat*, and is assigned the symbol J.

Since energy can neither be created nor destroyed, it is possible to make an energy balance for any given process. The amount of energy entering any region must equal the amount of energy leaving the region plus any energy stored within the region. Such an energy balance may be expressed as an energy equation. We shall first draw such a balance for a continuous flow process, and then modify the resultant equation for a nonflow process.

Figure 1 represents any steady flow device, such as a pump, a boiler, piping, etc. Fluid is entering at section 1 with a velocity U_1 and leaving at section 2 with a velocity U_2. At every point the conditions of steady flow are met:

1. The weight of fluid passing a given section in unit time is constant. No fluid storage occurs.
2. The fluid completely fills the passage.
3. All forms of energy entering or leaving the system do so at a constant rate.
4. The properties of the fluid (pressure, volume, and temperature) at any one section remain constant, though there may be a considerable variation from point to point through the device.

For convenience, the equation will be written for *1 pound of fluid passing* a given point in the device. This pound of fluid brings in at plane 1, relative to the datum plane:

Potential energy $= 1$ lb. x Z_1 ft. $= Z_1$ ft.-lbs.
Kinetic energy $= U_1^2/64.4.$ ft.-lbs.
Internal energy $= E_1$ B. t. u./lb. x J ft.-lbs./B. t. u.
$= JE_1$ ft.-lbs.
and Flow work $= P_1$ lbs./ft.^{2} x V_1 ft.$^3 = P_1V_1$ ft.-lbs.

Between planes 1 and 2:

Heat $= Q$ B. t. u. x J ft.-lbs./B. t. u.
$= JQ$ ft.-lbs.
and Work $= W$ ft.-lbs.
are added.

At plane 2, potential energy Z_2 foot-pounds, kinetic energy $U_2^2/64.4$ foot-pounds, internal energy JE_2 foot-pounds, and flow work P_2V_2 foot-pounds leave the system.

The complete equation, then, is:

$$Z_1 + U_1^2/64.4 + JE_1 + P_1V_1 + Q + W = Z_2 + U_2^2/64.4 + JE_2 + P_2V_2 \text{ ft.-lbs./lb. of fluid flowing.}$$

In the equation as written, heat and work put *into* the device are considered positive.

With the exception of the flow work term, the equation is simply a summary of definitions just discussed. Consider the effect of placing a diaphragm in the entering pipe at plane 1. If the fluid to the left of the diaphragm was removed, the system would be unchanged to the right providing the diaphragm was moved at a steady rate to the right so that steady flow was maintained. The force exerted on the diaphragm would be equal to the fluid pressure times the area of the channel, or P_1A_1, and for each pound of fluid. A_1L_1=cubic foot/pound of fluid, or V_1, the *specific volume* of the fluid at plane 1, and the work input per pound of fluid would be P_1V_1 foot-pounds/pound of fluid flowing. This is known as the *flow work* and is always present in a steady flow process.

Experience has shown that the PV product and the internal energy E are dependent only on the temperature. Since they appear together in the flow equation, $JE+PV$ are combined into a new term, JH, and H is called *enthalpy*. Like internal energy, it is expressed in B. t. u./pound. Enthalpy has no physical significance and may be used *only in case steady flow is realized.*

For a nonflow process, equation (1) simplifies to

$$JE_1 + Q + W = JE_2, \text{ or}$$

$$Q + W = J(E_2 - E_1) \text{ ft.-lb/lb. fluid} \quad \text{(2)}$$

The second law of thermodynamics states that energy will never of its own accord flow from a low energy level to a higher one.

This is best illustrated by examples. If we have a tank full of water on top of a hill and a pipe leading down the hill and into a pond, we will always find that if we open the valve in the pipe that the water will flow down from the tank and into the pond. There has never been an authentic case recorded of the water running out of the pond and up the hill of its own accord.

If we drop a piece of hot iron into a bucket of water, the water always gets warm and the iron is cooled to the temperature of the water because the heat flows from the higher temperature level in the iron to the lower temperature level of the water. It has never been found that the water froze and liberated heat to melt the iron instead of the iron being cooled.

2. PERFECT GASES

THE PERFECT GAS LAW

By physical experiments it has been shown that gases with monatomic and diatomic molecules, like Helium and Hydrogen, which are at a temperature considerably above the boiling point of the compound, obey certain laws to a good degree of approximation. These rules are known as the gas laws, and a gas which obeyed them exactly would be known as a "perfect gas."

Boyle found that *at constant temperature* the volume of a given quantity (weight) of a gas varies inversely as the pressure

$$\text{Ⓣ}\ \frac{V_2}{V_1}=\frac{P_1}{P_2}$$

or $P_1V_1=P_2V_2=$a constant (for the particular gas at the given temperature).

The symbol Ⓣ indicates that the temperature is maintained constant, and that the equation is true provided that this condition is met.

Charles found that *at constant volume* the pressure of a given amount of gas varies as a lineal function of the temperature. This relation is shown by figure 1. It has been found that if the absolute pressure is one unit at 32° F. (the temperature of melting ice at normal atmospheric pressure) then heating the gas without change of volume to 212° F. (the temperature of saturated steam at normal atmospheric pressure) will increase the pressure of the gas to 1.3662 units. If the straight line is extended until it cuts the zero pressure line, it will be found that this occurs 492° F. below the ice point, or at −460° F.

$$\frac{T_1+180}{T_1}=\frac{1.3662}{1.0000}$$

$$T_1=492°\text{ F. above absolute zero}$$

This temperature of −460° F. is known as absolute zero. A special scale of temperature using this as the zero temperature and having degrees of the same value as the Fahrenheit scale is used in engineering calculations. It is named the Rankine scale.

$$°\text{R.}=°\text{F.}+460$$

The absolute temperature usually used in physics has degrees of the same value as the centigrade degree. (1° C.=1.80° F.) Since the zero of the centigrade scale is the ice point (32° F.), absolute zero is at −273° C. This temperature scale is known as the Kelvin scale.

Absolute zero is a definite temperature, in a physical sense, and is considered to be the temperature at which all molecular activity ceases. It has never been reached experimentally, but it has been approached within a fraction of a °C.

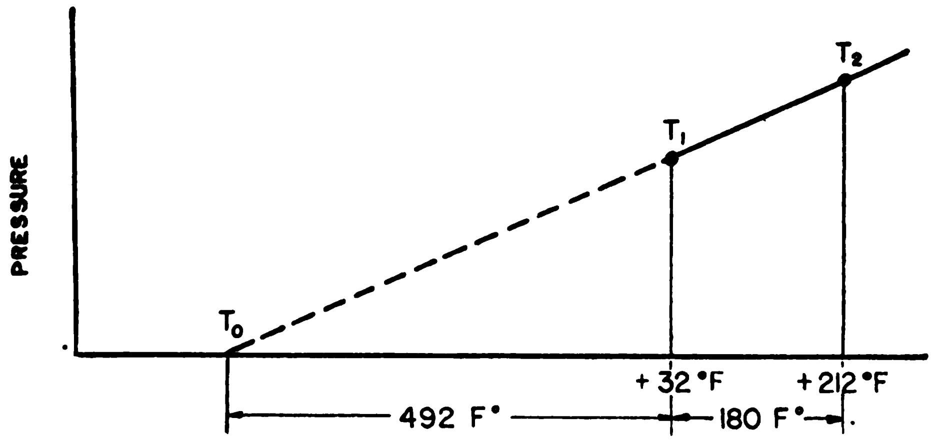


Figure 1.

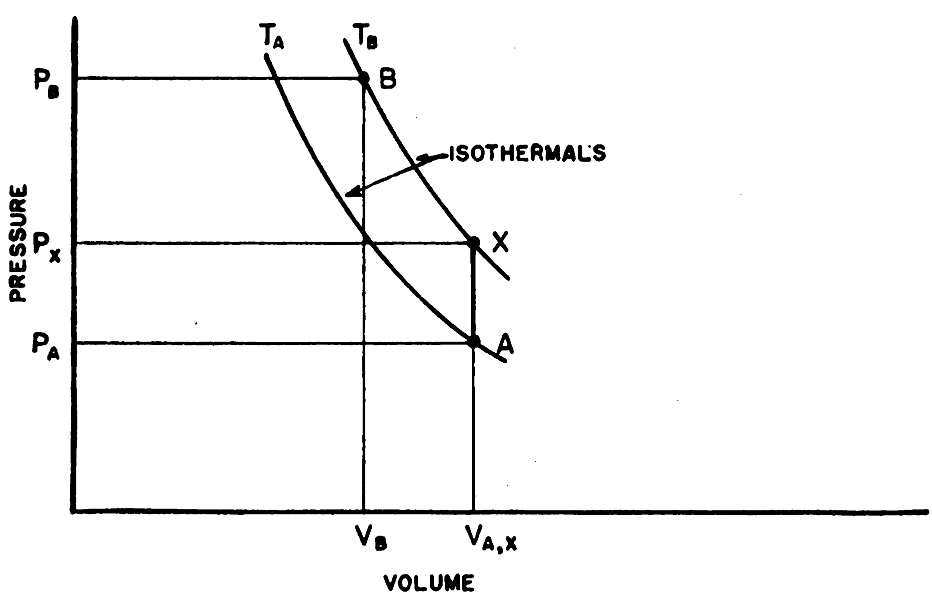


Figure 2.

Providing that the absolute (Rankine) temperature scale is used, Charles' law may be stated "at *constant volume* the absolute pressure varies directly as the absolute temperature." Expressed mathematically

$$\text{(V)}\ \frac{P_1}{P_2}=\frac{T_1}{T_2}$$

Charles also found that "*at constant pressure* the volume varies directly as the absolute temperature."

$$\text{(P)}\ \frac{V_1}{V_2}=\frac{T_1}{T_2}$$

For convenience it would be well to combine Boyle's law and the two forms of Charles' law into a single expression. This may be done as follows:

Take any two perfectly general states of a perfect gas with P, V, and T all different. These may be indicated as points A and B on a pressure-volume (P–V) diagram (fig. 2). The final state at any given point will depend only on the pressure, volume, and temperature, so it is possible to go from point A to point B by any convenient path, and any conclusions reached will be valid for any other path.

Through point A construct a line of constant temperature (from Boyle's law, this will be a line along which $PV=P_AV_A=$a constant) and through point B construct another line of constant temperature.

If heat is added to the gas at constant volume until the temperature is raised from T_A to T_B, point X on the diagram will be reached. By Charles' law

$$\text{(V)}\ \frac{P_X}{P_A}=\frac{T_X}{T_A}=\frac{T_B}{T_A} \qquad P_X=P_A\frac{T_B}{T_A}$$

The gas may now be compressed at constant temperature to P_B and V_B. By Boyle's law.

$$\text{(T)}\qquad \frac{P_X}{P_B}=\frac{V_B}{V_X}=\frac{V_B}{V_A} \qquad P_X=P_B\frac{V_B}{V_A}$$

Then

$$P_X=P_A\frac{T_B}{T_A}=P_B\frac{V_B}{V_A}$$

and $$\frac{P_AV_A}{T_A}=\frac{P_BV_B}{T_B}\ \text{a constant for a given}$$

quantity (weight) of the particular gas.

This expression is correct for any conditions of P, V, and T, because no limitations were set before deriving it.

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In engineering computations it is convenient to use a constant for *1 pound of a gas* and to use a specific constant for each real gas based on experimental data. This constant is represented by the symbol R and is known as the "gas constant."

This is a standard nomenclature used in some texts on engineering thermodynamics. Care should be taken to distinguish between R, the gas constant, and °R., the Rankine temperature. The perfect gas law may be expressed:

$$PV=RT$$

where P=absolute pressure in *pounds per square foot*, V=specific volume (volume occupied by 1 pound of gas) in *cubic feet*, T=absolute temperature in °R., and R is the gas constant in consistent units.

Values of the gas constant

Gas:	R
Air	53.3
Carbon dioxide	34.9
Carbon monoxide	55.1
Hydrogen	767.0
Nitrogen	55.1
Oxygen	48.3

The product of molecular weight × R=approximately 1,545 for all gases.

Normal atmospheric pressure at sea level is 14.7 pounds per square inch. Absolute pressure equals gage pressure plus atmospheric pressure.

Example 1.—A 40-cubic foot starting air flask for a Diesel engine contains air at 87° F. and at 750 pounds per square inch gage. How many pounds of air are in the flask?

$$PV=RT$$

$P=(750 \text{ lbs./in. ga.}+14.7 \text{ lbs./in.}^2)(144 \text{ in.}^2/\text{ft.}^2)$

$=764.7 \text{ x } 144=110{,}100$ *lbs./ft.² absolute*

V=cu. ft. air/lb. air

$R=53.3$

$T=87°\text{ F.}+460°\text{ F.}=547°\text{ R.}$ absolute temperature

$$V=\frac{53.3 \text{ x } 547}{110{,}100}=0.265 \text{ cu. ft.}$$

The flask contains

$$\frac{40}{0.265}=151 \text{ lbs. of air.}$$

SPECIFIC HEAT

The *specific heat* of a substance is the amount of heat which must be added or removed to change the temperature one degree Fahrenheit under stated conditions (i. e. along some given process line).

If a pound of gas contained within a cylinder with a piston locked in place is heated 1° F., the quantity of heat transferred to the gas is equal to the *specific heat at constant volume for that gas.* All of this heat is stored as internal energy of the gas.

If the same gas was heated through the same temperature range with the piston unlocked and moved against a resisting force at such a rate that constant pressure was maintained, then the amount of heat transferred would equal the *specific heat at constant pressure.* Moving the piston against a constant resisting force required that extra energy be supplied in addition to that necessary for the increase in internal energy of the gas. Hence the specific heat of a gas at constant pressure is always greater than the specific heat at constant volume.

There is a specific heat for every possible process. Only those for constant volume and constant pressure are commonly used.

Mean values of specific heats of gases from 32° F. to 400° F.

Gas	Specific heat at constant volume	Specific heat at constant pressure	$\frac{C_p}{C_v}=k$
	C_v	C_p	k
Air	0.173	0.241	1.40
Carbon dioxide	.162	.208	1.28
Carbon monoxide	.178	.249	1.40
Hydrogen	2.44	3.43	1.40
Nitrogen	.178	.249	1.40
Oxygen	.156	.218	1.40

Example.—14 pounds of air are heated at constant volume from 80° F. to 110° F., and then at constant pressure from 110° F. to 150° F. How much heat is required?

$$Q=14[(110-80)0.173+(150-110)0.241]$$
$$=14\ (5.19+9.64)=14\times 14.83=207.6 \text{ B. t. u.}$$

PROCESSES

A *process* is a chain of events during which a system, originally in one fixed state, undergoes certain energy transformations and thence arrives at a second fixed state.

> The process defines the path between fixed state points. By heating a gas in a closed container an increase in temperature and in pressure results without change in volume, and the process is defined as a *constant volume process.*

Five kinds of processes are normally recognized and used, partly because they satisfy conditions met with in actual practice, but more particularly because they are adapted to exact mathematical expression. They apply to processes using gases, liquids, or solids as the working substance, but in our present study we are interested primarily in gases.

(*a*) *Constant volume.*—A process conducted without changing the volume of the system.

$$\text{Ⓥ} \quad \Delta V=O \qquad PV=\text{a constant x } P$$

$$\text{Charles' law } \frac{P_1}{P_2}=\frac{T_1}{T_2} \text{ applies, as well as}$$

$$PV=RT$$

Specific heat $=C_v$

(*b*) *Constant pressure.*—A process conducted without changing the pressure of the system.

$$\text{Ⓟ} \quad \Delta P=O \qquad PV=\text{a constant} \times V$$

$$\text{Charles' law } \frac{V_1}{V_2}=\frac{T_1}{T_2} \text{ applies, as well as}$$

$$PV=RT$$

Specific heat $=C_p$

(*c*) *Constant temperature.* A process conducted without changing the temperature of the system.

$$\text{Ⓣ} \quad \Delta T=O \qquad PV=\text{a constant}=RT$$

Specific heat $=\infty$ (infinity)

(*d*) *Adiabatic.* A process conducted without transfer of energy as heat to or from the system.

$$Q=O \qquad P\cdot V^k=\text{a constant, as well as}$$

$$PV=RT$$

$$k=\frac{\text{Specific heat at constant pressure}}{\text{Specific heat at constant volume}}=\frac{C_p}{C_v}$$

Specific heat $=O$

(*e*) *Polytropic.*—A process conducted with a constant specific heat so that—

$$PV^n=\text{a constant, as well as}$$

$$PV=\text{RT}$$

$n=$ an empirical exponent.

$$\text{Specific heat}=C_v\frac{n-k}{n-1}$$

The polytropic is the most artificial of the five processes and is chosen primarily because it is simple to treat mathematically. Many real processes, such as the compression of a gas in a water cooled cylinder, are neither isothermal nor adiabatic. Experi-

ments have shown that the formulae above closely approximate the real conditions encountered, providing that a suitable value of n is selected.

Example 3. 17 cubic feet of air at 10 pounds per inch gage is compressed to 45 pounds per inch gage. The process is polytropic with $n=1.37$. What will be the final volume?

$$PV^{1.37}=\text{constant}$$

P=absolute pressure (any units in this case because the problem is to solve a ratio)

V=volume (any units)

$$P_1V_1^{1.37}=P_2V_2^{1.37}$$

$$V_2=V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{1.37}}$$

$$=17\left(\frac{10+14.7}{45+14.7}\right)^{0.730}$$

$$=17(0.413)^{0.730}$$

$$=17(0.525)=8.92 \text{ cu. ft.}$$

NOTE: a^y=antilog y log a. (Antilog: number whose log is . . .)

$$(0.413)^{0.730}=\text{antilog } 0.730 \log 0.413$$
$$=\text{antilog } 0.730\ (-1.000+0.616)$$
$$=\text{antilog}-0.280$$
$$=\text{antilog }(-1.000+0.720)$$
$$=0.525$$

Example 4.—3 pounds of nitrogen occupying 40 cubic feet at 85° F. are compressed adiabatically to a volume of 12 cubic feet. What is the pressure immediately after compression?

$$P_1V_1=RT_1$$

$$P_1=\frac{55.1\cdot 545}{13.33}=2250 \text{ lb./ft.}^2\text{ absolute}$$

$$P_1V_1^{1.41}=P_2V_2^{1.41}$$

$$P_2=2250\left(\frac{40}{12}\right)^{1.41}$$

$$=2250\cdot 5.45$$

$$=12{,}250 \text{ lb./ft.}^2\text{ absolute}$$

or 85.1 lb./in.^{2} absolute

Note: $\left(\frac{40}{12}\right)^{1.41}=(3.33)^{1.41}=\text{antilog } 1.41 \log 3.33$

$$=\text{antilog } 1.41\cdot 0.522$$
$$=\text{antilog } 0.736$$
$$=5.45$$

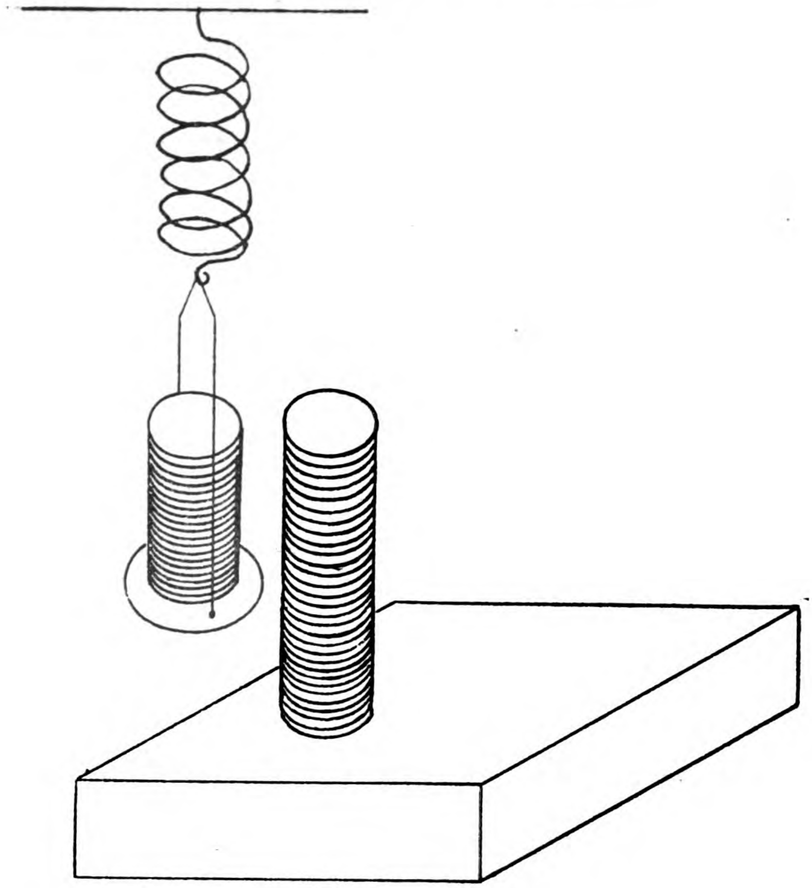


Figure 3.

Problems in fractional exponents may be solved in the foregoing manner using the logarithm scale on a slide rule. This is the L scale on most rules (a 10-inch scale divided into 1,000 equal parts) and is matched with a 10-inch scale, usually the C or D scale. The number read from the L scale is the mantissa of the logarithm. This may be checked, remembering that the log of 2 is approximately 0.30 and the log of 5 is approximately 0.70.

Fractional powers may be worked directly with log-log slide rule scales.

REVERSIBLE PROCESSES

A reversible process is one during which a system may change from one fixed state to another, and then be returned to the original state along the identical process line. At the completion of the

process, the system and all related surroundings must have returned to their initial states.

Reversibility cannot be achieved if at any time there is a sensible temperature difference across a boundary where heat is being transferred, or if there is mechanical or fluid friction.

The principle of reversibility may be visualized by considering a system consisting of a small tray suspended from a perfectly elastic spring and hanging beside a pile of dimes (fig. 3). If the spring constant is two dime thicknesses for each dime's-weight increase in load, then we can move each dime horizontally from the pile to a stack on the tray. The process consists of transforming potential energy of the dimes' original positions to potential energy of deformation of the spring. By moving the dimes horizontally back from the stack on the tray to the main pile, we can return the system to its original state. If no effort is required to move the dimes back and forth, then this is an example of a reversible process. However, if moving dimes makes the operator tired, then the related surroundings have not returned to their initial states, and the process is irreversible.

3. WORK

Work has already been defined as mechanical energy in transition. The unit of work is the foot-pound, and the common unit for the *rate* of doing work is the horsepower.

The amount of work required to move a distance of 20 feet against a resistance of 10 pounds is 200 foot-pounds. Consider a piston with an area of A square feet being pushed against a pressure of P pounds per square foot (fig. 1). The total force on the piston will be PA pounds. In pushing the piston ΔL feet, PA (ΔL) foot-pounds of work will be required. Nothing that A (ΔL) is an increment of volume and may be expressed as ΔV, then $W=P(\Delta V)$. This is true, providing that P does not change sensibly as the piston is moved a distance ΔL, and providing that the increment of ΔV *is* small enough, this will be accurate regardless of the variation of P as V is changed. For a considerable change of pressure, where n increments of ΔV and corresponding values of P are sufficient to give the required accuracy, then

$$W=P_1(\Delta V)_1+P_2(\Delta V)_2+P_3(\Delta V)_3\text{---------------------}P_n(\Delta V)_n$$

or $W= \quad P\,dV$, using the notation of calculus

Whenever an integration is performed in calculus, a constant of integration is acquired. For instance, $\int x^2\,dx=\frac{1}{3}x^3+C$. If the integration is performed between definite limits, the constant

is cancelled. For example, suppose the above expression was integrated between limits of 2 and 3, then

$$\int_2^3 x^2dx = [1/3x^3 + C]_2^3$$
$$= (27/3 + C) - (8/3 + C) = 19/3$$

In thermodynamics we are invariably interested in changes (i. e., we are always working between definite limits) and hence the constants are neglected when integrations are performed.

We are now in a position to derive expressions for nonflow work for perfect gases for the five common kinds of processes already described. In accordance with the convention used with the flow equation, work put into the gas is considered positive and work obtained is considered negative.

In all cases *complete reversibility is assumed.*

1. For a constant volume process,

$$W = P\triangle V, \text{ but } \triangle V = O, \text{ hence}$$
$$W = O$$

2. For a constant pressure process

$$W = -\int_{V_1}^{V_2} P\, dV \text{ and since } P = \text{a constant,}$$
$$= -P\int_{V_1}^{V_2} dV$$
$$= -P(V_2 - V_1) = +P(V_1 - V_2)$$

3. For a constant temperature process

$$W = -\int_{T_1}^{V_2} P\, dV$$

From Charles' law $PV = RT$, or

$$P = \frac{RT}{V}$$

$$W = -RT\int_{V_1}^{V_2}\frac{dV}{V} = -RT\, Ln\frac{V_2}{V_1} = +RT\, Ln\frac{V_1}{V_2}$$

or P_1V_1 in $\frac{V_1}{V_2}$, or $P_2V_2\, Ln\frac{V_1}{V_2}$

[NOTE: *Ln* is an abbreviation for natural logarithm, and is also written $\log_e$.]

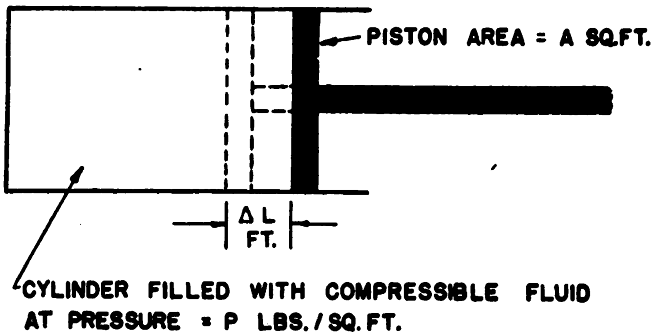


Figure 1.

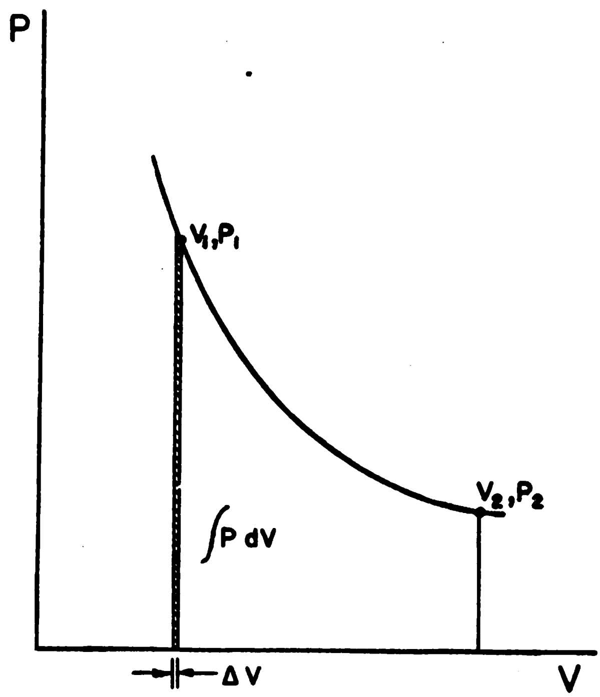


Figure 2.

4. For an adiabatic process

$$W=-\int_{V_1}^{V_2} P\,dV$$

$$PV^k=C$$

$$W=-C\int_{V_1}^{V_2} V^{-k}dV=-\frac{CV_2^{1-k}-CV_1^{1-k}}{1-k}$$

but $C=P_1V_1^k=P_2V_2^k$

$$W=-\frac{P_2V_2+P_1V_1}{1-k}=\frac{P_2V_2-P_1V_1}{k-1}, \text{ or}$$

$$W=\frac{R}{k-1}\,T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}-1\right]$$

Because $P_1V_1^k=P_2V_2$

$$V_2=V_1\left(\frac{P_1}{P_2}\right)^{\frac{1}{k}}=V_1\left(\frac{P_2}{P_1}\right)^{-\frac{1}{k}}=\frac{RT_1}{P_1}\left(\frac{P_2}{P_1}\right)^{-\frac{1}{k}}$$

Whence $P_2V_2=RT_1\left(\frac{P_2}{P_1}\right)\left(\frac{P_2}{P_1}\right)^{-\frac{1}{k}}=RT_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$

Then $W=\frac{P_2V_2-P_1V_1}{k-1}=\frac{RT_1}{k-1}\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}-\frac{RT_1}{k-1}$

5. For a polytropic process

$$W=\frac{P_2V_2-P_1V_1}{n-1}=\frac{R}{n-1}T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}-1\right]$$

WORK ON THE P–V DIAGRAM

For a small change in volume the work increment is $P(\Delta V)$. Represented on P–V coordinates, this is the area of a narrow strip of width ΔV extending from the volume axis to the process line, as indicated in the sketch, figure 2. The total amount of work available from a reversible expansion from P_1 to P_2 along the process line in the figure would be the sum of all of the work increments, or the area under the process line and above the volume axis between the volume limits V_1 and V_2.

Logarithms to base 10

No.	0	1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962

Logarithms to base 10—Continued

No.	0	1	2	3	4	5	6	7	8	9
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538

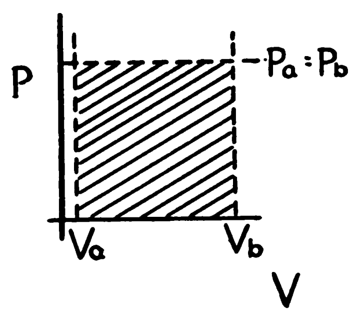
Logarithms to base 10—Continued

No.	0	1	2	3	4	5	6	7	8	9
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996
100	0000	0004	0009	0013	0017	0022	0026	0030	0035	0039

The utility of this relation will become apparent when indicator diagrams are considered. Indicator cards are experimental P–V diagrams of actual engine performance obtained directly from a running engine, and by measurement of the area under the process lines, it is possible to calculate the work developed by the engine.

SAMPLE PROBLEMS

1. A gas initially in state "a" ($P_a=600$ pounds per square inch absolute, $V_a=0.3$ cubic foot) is heated at constant pressure until state "b" is reached, where $V_b=0.6$ cubic foot. What work was done on the gas during the process?



$$\begin{aligned}\text{Work} &= P_a(V_b - V_a)\\ &= 600\times 144\times(0.6-0.3)\\ &= 25920 \text{ ft.-lbs.}\end{aligned}$$

2. Ten pounds of air are contained in a receiver at a temperature of 55° F., and a pressure of 100 *pounds per square inch absolute.* Air leaks out until at a later time the pressure is found to be only 40 *pounds per square inch absolute* at a temperature of 50° F. What weight of air leaked out?

$$PV = WRT \text{ (for } W \text{ lbs. of fluid)}$$

$$R_1 = R_2 = \frac{P_1V_1}{W_1T_1} = \frac{100V_1}{10\times 515} = \frac{V_1}{51.5}$$

$$V_1 = V_2\,;\ 40\times V_2 = W\frac{V_1}{51.5}510$$

$W=4.04$ lbs., wt. of air still in receiver

$10.00-4.04=5.96$ lbs., *answer*

3. The lubricating oil used in an air compressor has a flash point of 400° F., but it is desirable the temperature of the air be kept at least 50° F. below the flash point of the oil. The compressor draws air in at 14.7 pounds per square inch absolute, and a temperature of 80° F. If the compression is isentropic, find the maximum allowable compression pressure.

$T_1=80°$ F. $=540°$ R.
$T_2=350°$ F. $=810°$ R.
$P_1=14.7\times 144=2130$ p. s. f.
$k_{air}=1.40$

$$\frac{P_1}{P_2}=\left(\frac{T_1}{T_2}\right)^{\frac{k}{k-1}};\frac{P_2}{P_1}=\left(\frac{T_2}{T_1}\right)^{\frac{k}{k-1}}$$

$$\frac{P_2}{14.7}=4.140$$

$P_2=61.0$ p. s. i. abs., max. allowable comp. press.

$$\frac{840}{540}=\frac{T_2}{T_1}=1.5$$

$$\frac{k}{k-1}=\frac{1.40}{0.40}=3.5$$

$(1.5)^{3.5}=N$
Log $N=3.5$ log 1.5
$=3.5\times 0.1762$
$=0.6170$
$N=4.140$

4. CYCLES

Cycles.—A series of processes which may be repeated over and over constitutes a *cycle.* The function of a cycle is to provide a means of transferring heat from a high temperature level to a lower temperature level, extracting energy as work (or the converse, to provide a means of transferring heat from a low temperature level to a higher one by providing external energy as work). A cycle may be operated so that the same pound of fluid goes through the series of processes and returns to its original state. This is a *steady flow cycle* and is exemplified by a steam power plant. A second type of cycle may be operated so that the machine and the working substance reach the same condition in each successive cycle, but using a fresh supply of working fluid each cycle. This is an *intermittent cycle* and is exemplified by an internal combustion engine. A third type of cycle is similar to the second, except that the working substance is contained within a cylinder fitted with a piston and the machine and working fluid go through the cycle together, the same supply of fluid being used repeatedly. This is a *nonflow* process. No modern machines operate on nonflow processes, but they lend themselves to analyses which are valid in a practical degree for engines working on cycles of the second class.

No cycle can be more efficient than a reversible cycle, and all reversible cycles are equally efficient if operated between the same temperature limits. A reversible cycle is, of course, more efficient than a similar irreversible cycle because in the latter case energy is wasted. A reversible cycle which yielded 100 units of energy per pound of working fluid cooled from the high to the low temperature could

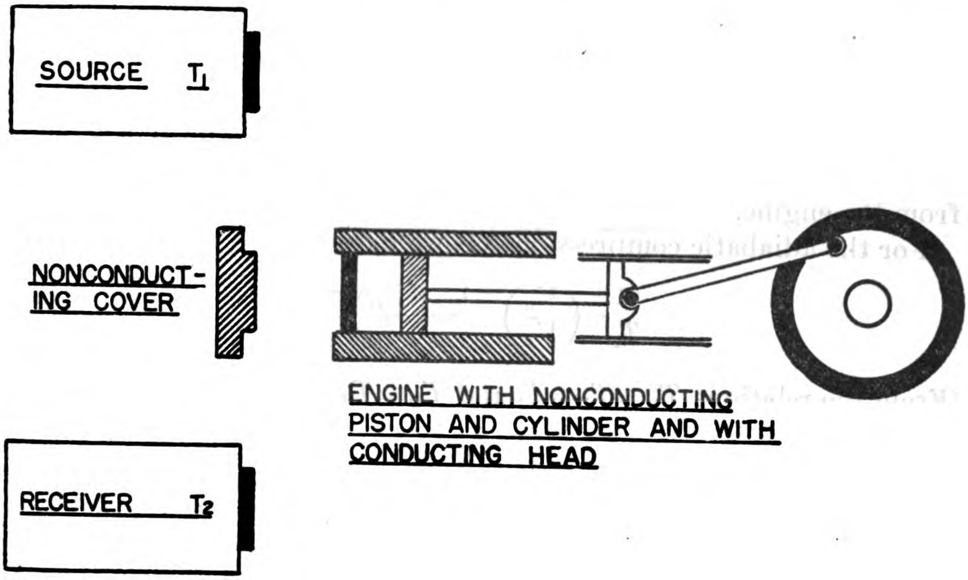
return a pound of the working fluid to the higher temperature from the lower at the expense of 100 units of energy. If another more efficient machine was available which would yield 120 units of energy per pound of working fluid cooled, then this machine could be used to drive the less efficient engine to return the working fluid to the original temperature and still yield 20 units of net energy. This would amount to creating energy in violation of the first law of thermodynamics (which is always rigidly in force). It therefore follows that all reversible engines are equally efficient when operating between the same temperature limits.

Since all reversible cycles are equally efficient, conclusions drawn from the study of any particular cycle are valid for any conceivable cycle. The classic academic cycle is the *Carnot cycle* because it is readily analyzed and even though it is eminently impractical the results of the analysis are applicable to practical cycles.

Figure 1 is a diagram of an engine to operate on the Carnot cycle. The Carnot cycle is a nonflow cycle. The working medium is a compressible fluid—a perfect gas. It is assumed that this perfect gas has the physical constants possessed by air at room temperature. One pound of this gas is contained in a cylinder of special construction; the walls and piston are incapable of conducting heat, but the cylinder head is a perfect conductor. An infinite supply of heat at some definite constant temperature is stored in a container with a perfectly conducting wall which will cover the piston head when so installed. This is termed the constant temperature source.

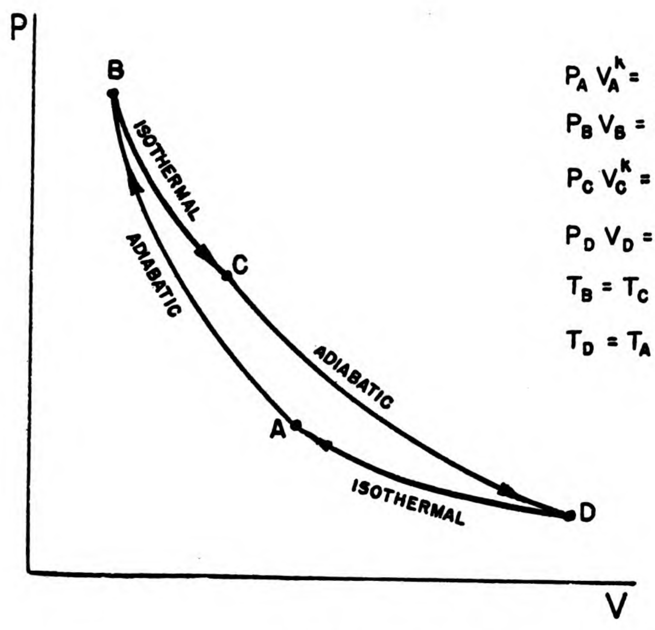
A container of infinite capacity for heat storage at constant temperature is provided into which the heat to be rejected by the engine may be transferred through a perfectly conducting wall which, like that in the source, will cover the cylinder head. This is termed the constant temperature receiver. The engine is also provided with an insulating cover which can be installed over the cylinder head and which is incapable of conducting heat.

The engine is operated on the following cycle. With the working substance initially at the receiver temperature, the nonconducting cover is installed over the cylinder head, and the gas is adiabatically and reversibly compressed until it is heated by the compression to the source temperature. The cover is then removed and the cylinder head installed in contact with the source. A quantity of heat is taken from the source, the piston being moved so that the transfer is isothermal and reversible. After detaching the cylinder head from the source and reinstalling the cover, the gas is allowed to expand adiabatically and reversibly to the receiver temperature. Then the cover is removed, the cylinder head installed in contact with the receiver, and heat is rejected isothermally and reversibly to the receiver until the initial state conditions of the working substance are



CARNOT ENGINE

Figure 1.



$$P_A V_A^k = P_B V_B^k \quad (k=1.41)$$

$$P_B V_B = P_C V_C = RT_1 \quad (R=53.3)$$

$$P_C V_C^k = P_D V_D^k$$

$$P_D V_D = P_A V_A = RT_2$$

$$T_B = T_C = T_1$$

$$T_D = T_A = T_2$$

CARNOT AIR CYCLE

Figure 2.

reached. During heat reception from the source and the subsequent expansion, work is obtained from the engine. During the heat rejection and the compression, work must be supplied to the engine. The P–V–T relations for the cycle are shown in figure 2. The area enclosed within the four process lines is proportional to the net work obtainable from the engine.

For the adiabatic compression AB we have

$$\frac{T_1}{T_2}=\left(\frac{V_A}{V_B}\right)^{k-1}=\left(\frac{V_D}{V_C}\right)^{k-1}$$

(From the relations $T_B=T_C=T_1$ and $T_D=T_A=T_2$ and $PV^k=$const.)

when

$$\frac{V_A}{V_B}=\frac{V_D}{V_C}$$

The heat received along BC is

$$Q_1=RT_1\log_e\frac{V_C}{V_B}$$

The heat rejected along DA is

$$Q_2=RT_2\log_e\frac{V_D}{V_A}$$

The efficiency of the cycle is

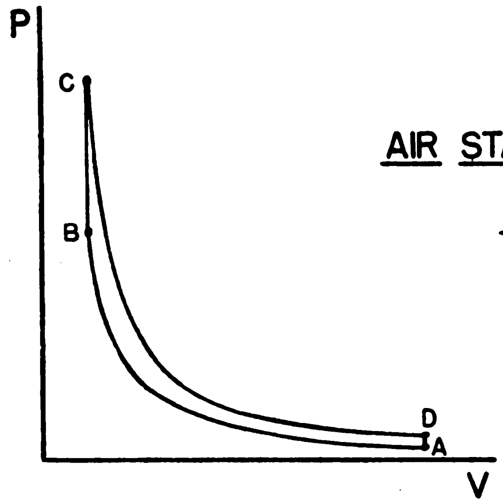
$$\frac{\text{Heat received}-\text{Heat rejected}}{\text{Heat received}}=\frac{Q_1-Q_2}{Q_1}$$

$$\frac{Q_1-Q_2}{Q_1}=\frac{RT_1\log_e\frac{V_C}{V_B}-RT_2\log_e\frac{V_D}{V_A}}{RT_1\log_e\frac{V_C}{V_B}}=\frac{T_1-T_2}{T_1}$$

This efficiency, $\frac{T_1-T_2}{T_1}$, is the limiting thermal efficiency for any conceivable cycle operating between the temperature limits T_1 and T_2.

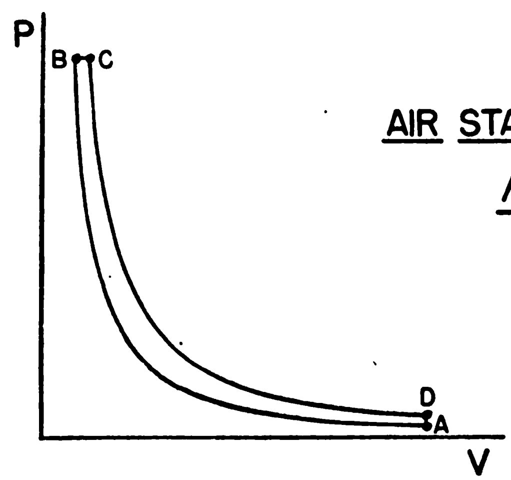
As stated before, the Carnot cycle is of theoretical interest only. The nonflow cycles which approach the practical intermittent cycles used in internal combustion engines are (1) the *Otto cycle*, which is an approximation of the working cycle of moderate compression spark-ignition engines, (2) the *Diesel cycle*, which is an approximation of the working cycle of a slow speed compression-ignition engine, and (3) the *Sabathé cycle*, which is an approximation of the working cycle of high speed compression-ignition engines and very high compression spark-ignition engines.

The working processes used in the Otto cycle are shown on P–V coordinates in figure 3. The working medium ("perfect air"), initially at state A is adiabatically compressed to state B. Heat is added at



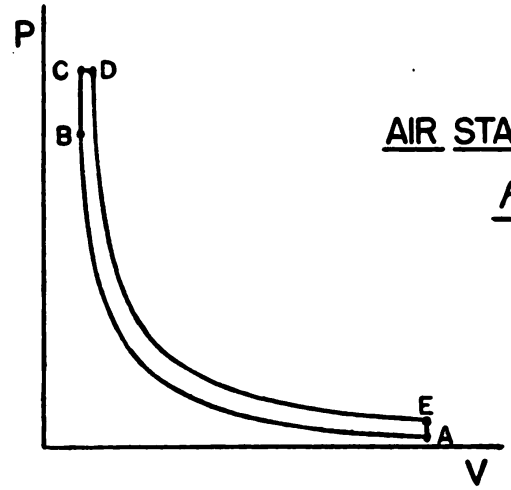
AIR STANDARD OTTO CYCLE

Figure 3.



AIR STANDARD DIESEL CYCLE

Figure 4.



AIR STANDARD SABATHÉ CYCLE

Figure 5.

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constant volume with rising temperature and pressure to C. Adiabatic expansion follows to state D. Constant volume rejection of heat with falling temperature and pressure returns the working medium to state A.

The compression ratio, r_c, is defined as the ratio of the volume of the working medium at state A to its volume at state B.

$$\text{Compression ratio} = r_c = \frac{V_A}{V_B} = \frac{V_D}{V_C}$$

By an analysis similar to that used for the Carnot cycle, it can be proven that the theoretical thermal efficiency is dependent on the compression ratio.

$$\text{Efficiency} = 1 - \frac{T_A}{T_B} = 1 - \left(\frac{1}{r_c}\right)^{k-1}$$

This is logical because the temperature at which combustion commences, T_B, is dependent on the compression ratio providing the initial temperature, T_A, is fixed.

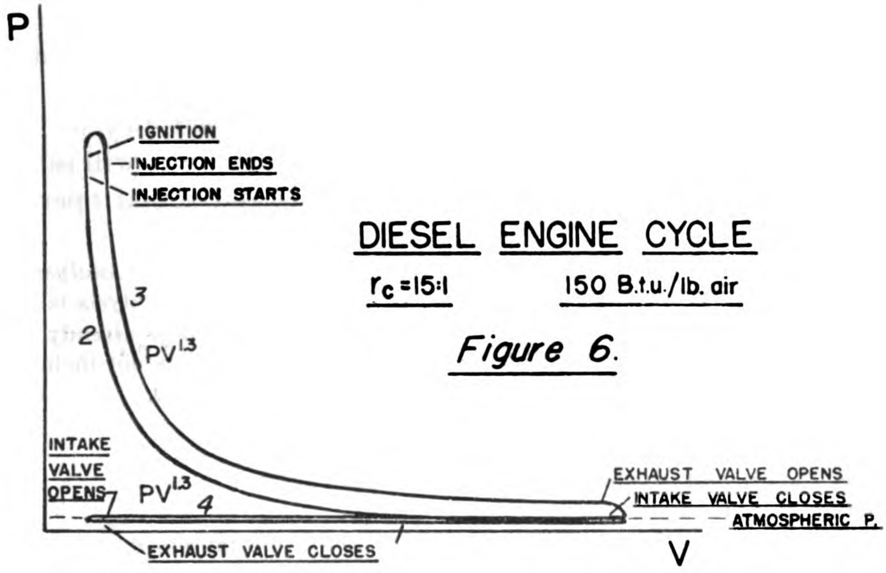
The working processes used in the Diesel cycle are shown on P–V coordinates in figure 4. The working medium, initially at state A, is adiabatically compressed to state B. Heat is then introduced at constant pressure to state C. Expansion adiabatically to state D follows. Constant volume rejection of heat with falling temperature and pressure returns the working medium to state A.

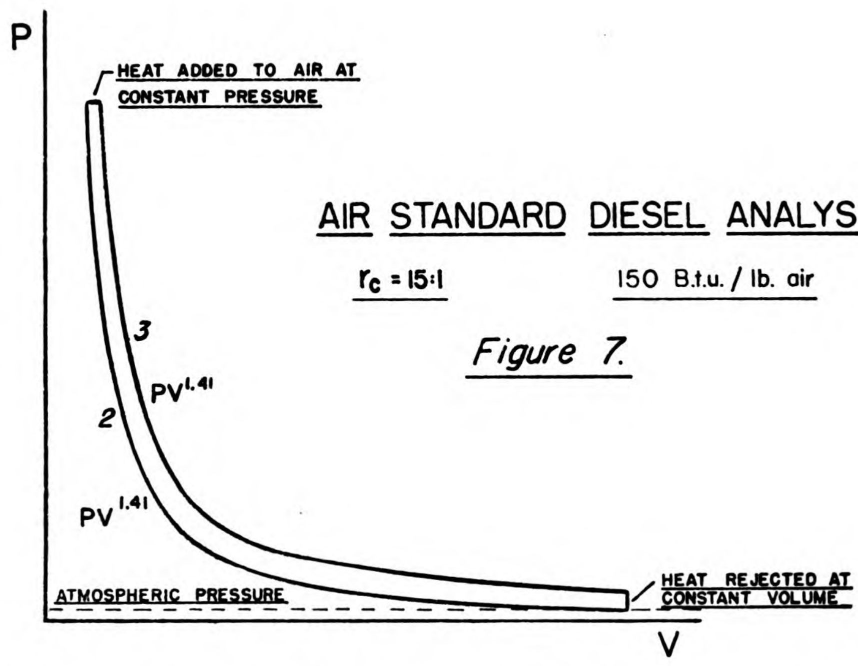
The *compression ratio*, r_c, is defined as for the Otto cycle as the ratio of the volume of the working medium at state A to its volume at state B. The *expansion ratio*, r_e, is defined as the ratio of the volume of the working medium at state D to its volume at state C. The theoretical thermal efficiency of the Diesel cycle can be proven to be.

$$\text{Efficiency} = 1 - \frac{T_D - T_A}{k(T_C - T_B)} = 1 - \frac{\left(\frac{r_c}{r_e}\right)^k - 1}{k(r_c)^{k-1}\left[\frac{r_c}{r_e} - 1\right]}$$

The working processes of the Sabathe cycle are shown on P–V coordinates in figure 5. The working medium, initially at state A, is adiabatically compressed to state B. Heat is then introduced at constant volume to C, and then at constant pressure to D. Adiabatic expansion to state E follows. Constant volume rejection of heat with falling temperature and pressure returns the working medium to state A. With the compression ratio defined as the ratio of volume A to volume B, and the expansion ratio defined as the ratio of volume E to D, the theoretical thermal efficiency can be proven to be

$$\text{Efficiency} = 1 - \frac{T_E - T_A}{T_C - T_B + k(T_D - T_C)}$$





COMPARISON OF THE AIR STANDARD EFFICIENCIES OF THE OTTO, DIESEL, AND SABATHÉ CYCLES

The cycle analyses discussed above are known as air standard analyses because they are based on cycles operating with air as the working medium. Heat is supplied from an external source. They are non-flow cycles. This is not the method of actual operation of internal

combustion engines, and the results obtained from the analyses in consequence do not coincide quantitatively with engine performance. The general principles derived, however, are valid.

A tabular comparison between the events in a real 4-cycle Diesel engine cycle and those in the air standard Diesel analysis will serve to emphasize the differences between the analysis and actual operating conditions.

Diesel Engine Cycle	*Air Standard Diesel Analysis*
1. On the first stroke of the cycle, air is drawn into the working cylinder. The flow of air into the cylinder occurs because the pressure in the engine cylinder is lower than the pressure in the intake manifold. Work is required to induct the air and is subtracted from the engine output.	1. The air standard analysis is a non-flow process. The air is already in the cylinder. This stroke is not included in the theoretical analysis.
2. The intake valve is closed and the air charge is compressed polytropically until fuel injection begins. $PV^{1.3}$ is approximately constant. Some combustion products are present from the proceeding stroke. Injection begins near the end of the compression stroke. First, some of the fuel vaporizes, lowering the air temperature. Then as the fuel reaches its self ignition temperature and the delay period is ended, ignition occurs. The fuel is oxidized, resulting in a change in the chemical composition of the charge, and raising the temperature and pressure rapidly while the fuel injected during the ignition delay is consumed. This occurs at about top dead center and more or less approaches a constant volume process.	2. The air is compressed adiabatically until the piston reaches the end of its stroke. $PV^{1.41}$ is constant.
3. Injection continues during the start of the down stroke of the piston. At first the rate of piston travel is slow so that the pressure continues to rise. As the piston motion accelerates the pressure reaches a maximum and starts to drop. When the amount of fuel required for the load on the engine is injected, injection is terminated. Burning continues as long as unburned fuel is present.	3. At the start of the down stroke, heat is transferred to the air in the cylinder from an external source. The rate of heat addition is controlled so that the pressure in the cylinder remains constant until all of the heat required for the "load" has been added.

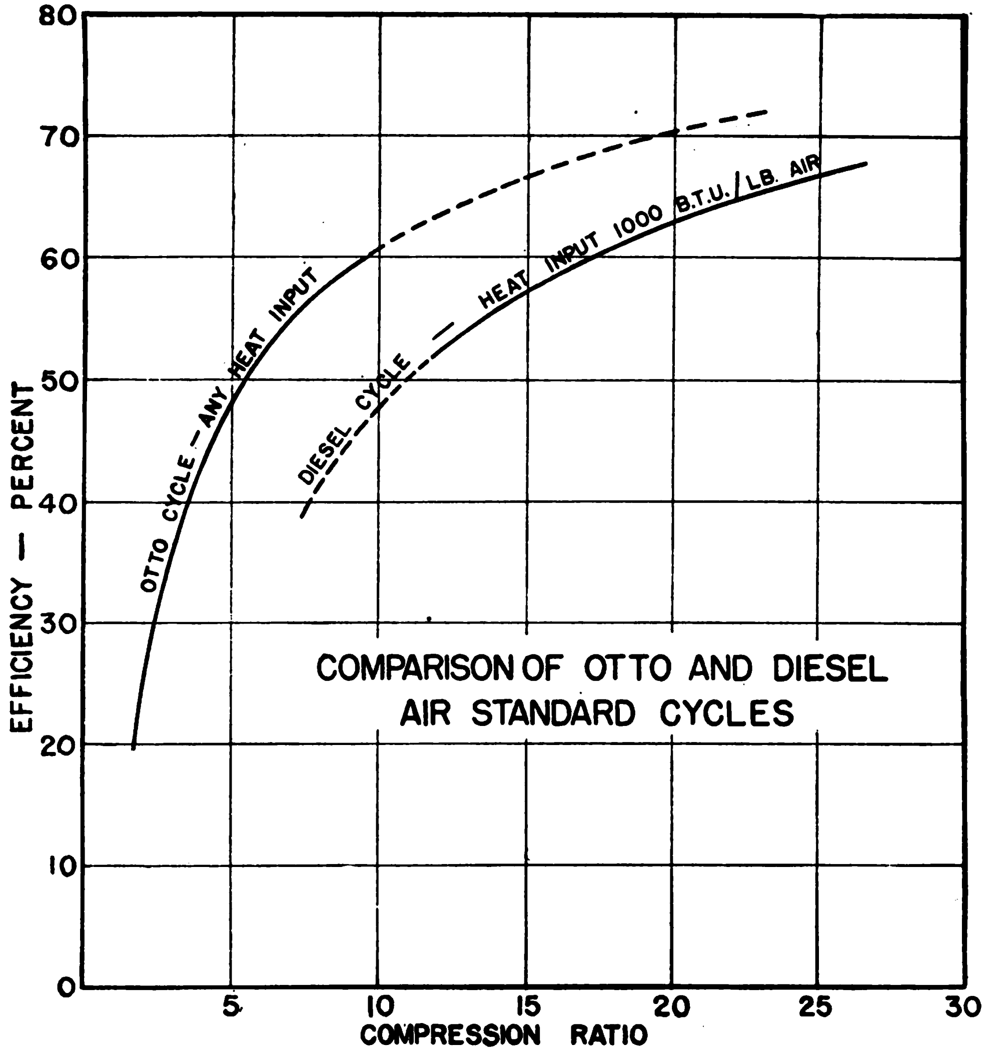


Figure 8.

The gaseous products of combustion, consisting of CO_2, CO, N_2, O_2, H_2O, and SO_2 (principally) are expanded. The expansion is not polytropic during the early part of the expansion owing to afterburning. Toward the end of the expansion, the rate of expansion approximates the polytropic $PV^{1.3}$=const.

Near the end of the stroke the exhaust valve opens, and the pressure in the cylinder drops along a falling pressure curve to a value slightly below the exhaust manifold pressure. This is due to the inertia effects of the rush of gas from the cylinder.

The air is expanded adiabatically. $PV^{1.41}$=const.

At the end of the stroke, heat is removed from the air at constant volume and is rejected to an external receiver which varies in temperature to accept it reversibly.

4. Most of the combustion products are exhausted into the exhaust manifold during the fourth stroke. Work is required to exhaust the combustion products and is subtracted from the engine output.

4. The air standard analysis is a nonflow process. There is no exhaust stroke included in the theoretical analysis.

For a given compression ratio, the Otto cycle is more efficient than the Diesel cycle, and the Sabathé cycle is more efficient than the Diesel cycle but less efficient than the Otto cycle. This is shown graphically in figure 8.

The maximum compression ratio for the Otto cycle is dependent (as will be discussed in a later assignment) on the tendency of the

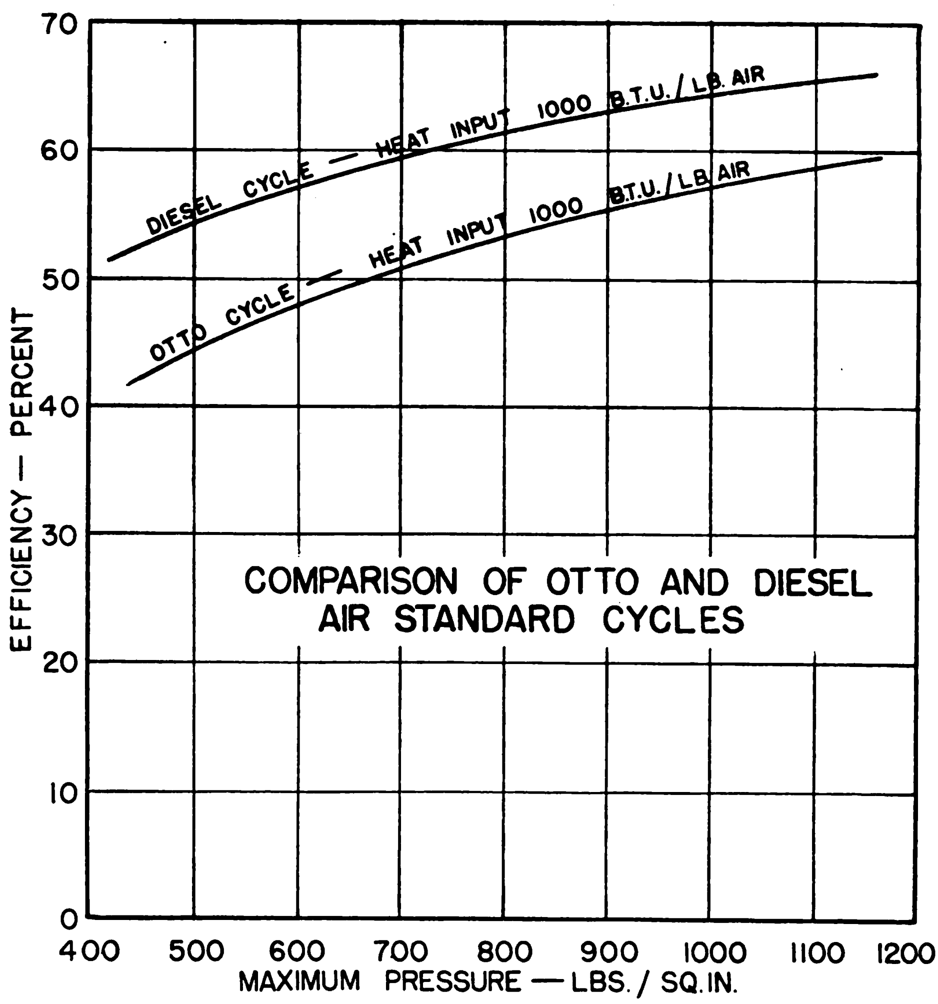


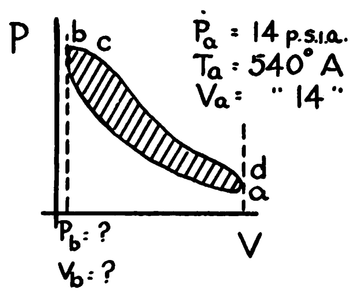
Figure 9.

gasoline used in a spark ignition engine to auto-ignite as the result of the temperature reached during rapid compression of the charge. The maximum compression ratio for a compression ignition engine is not limited. The limit, rather, is imposed by the maximum temperature and pressure which can be withstood by the engine. The best attainable thermal efficiency is therefore realized by compressing the air charge to the maximum allowable temperature and pressure and then burning the charge at this constant high pressure, operating the engine on the Diesel cycle. This is shown by figure 9.

The Sabathé cycle is strictly a compromise resorted to because it is impractical to operate a high-speed engine on a true Diesel cycle. Some modern special gasolines have a very high resistance to self-ignition and engine limitations require that they be operated on the Sabathé cycle too. Fuel technological advancement has thus made it possible to operate high-speed Diesel and gasoline engines with approximately equal thermal efficiencies and resolves the question to one of the relative merits of engine costs, mechanical simplicity and reliability, and fuel fire hazards.

SAMPLE PROBLEMS

1. A 4-stroke cycle Diesel engine has a 14:1 compression ratio. At the end of the suction stroke the air pressure is 14 pounds per square inch absolute, and the air temperature is 80° F. What will be the temperature and pressure at the end of the compression stroke?



$k=1.40$

$$\frac{T_b}{T_a}=\left(\frac{P_b}{P_a}\right)^{\frac{k-1}{k}}=\left(\frac{V_a}{V_b}\right)^{k-1}\cdot\frac{T_b}{540}=\left(\frac{14}{1}\right)^{0.40}$$

$$\frac{1550}{540}=\left(\frac{P_b}{14}\right)^{\frac{0.4}{1.4}}=2.87 \qquad T_b=1550° \text{ Abs.} \equiv 1090° \text{ F.}$$

$$\frac{P_b}{14}=(2.87)^{\frac{1.4}{0.4}}$$

$P_b=560$ p. s. i. abs.

2. What will be the *theoretical* thermal efficiency of a Diesel engine with a 16:1 compression ratio, assuming it could operate on the ideal Diesel cycle? Assume expansion ratio of 14:1. k=1.30.

$$\text{Eff.}=1-\frac{\left(\frac{r_c}{r_e}\right)^k-1}{k(r_c)^{k-1}\left[\frac{r_c}{r_e}-1\right]};\ \text{Eff.}=1-\frac{\left(\frac{16}{14}\right)^{1.3}-1}{1.3(16)^{0.3}\left[\frac{16}{14}-1\right]}$$

$$\left(\frac{16}{14}\right)^{1.3}=N_1$$

$$\begin{aligned}
\text{Log } N_1&=1.3 \log 1.14 & &=1-\frac{(1.14)^{1.3}-1}{1.3(16)^{0.3}[1.14-1]}\\
&=1.3\times.0569 & &=1-\frac{1.18-1}{1.3\times2.30\times0.14}\\
&=.0740 & &\\
N_2&=1.18 & &=1-\frac{0.18}{0.418}\\
16^{0.3}&=N_2 & &\\
\text{Log } N_2&=0.3 \log 16 & &=1-0.43\\
&=0.3\times1.2041 & \text{Eff.}&=57\%\\
&=.36123 & &\\
N_2&=2.30 & &
\end{aligned}$$

Chapter IV. FUNDAMENTALS

1. COMPONENT PARTS

An internal combustion (I. C.) engine is a machine for converting chemical energy to mechanical energy by burning a fuel with air in a confined space and expanding the products of combustion, extracting energy as work. An I. C. engine is a type of heat engine.

There are two forms of I. C. engines, reciprocating I. C. engines and I. C. turbines. At the present time reciprocating I. C. engines dominate the field, although there are a few semipractical I. C. turbines in existence and in experimental operation. It is probable that at some time in the future the I. C. turbine will supersede reciprocating I. C. engines, but this must await further advances in metallurgy. We shall therefore restrict this discussion to reciprocating machines.

The two principal species of heat engines in common use are steam engines and I. C. engines. In the case of the steam engine the chemical energy of fuel is changed to heat energy in a furnace and transferred to heat energy in steam in a boiler. The heat energy of the steam is partially converted to work in a steam engine. In the case of the I. C. engine combustion takes place in the engine itself, obviating the necessity of both the boiler and the water. As shown in the assignments on thermodynamics, the thermal efficiency of any heat engine is dependent on the temperature limits between which the engine operates. I. C. engines operate between greater temperature differences than any steam plants save the ultra high pressure installations now found in a few shore establishments. This enhanced efficiency coupled with relative simplicity of operation is rapidly making the I. C. engine dominate in units up to approximately 2,000 to 3,000 horsepower and plant installations up to about 5,000 or 10,000 horsepower. For greater power requirements, steam plants are capable of developing greater power within a given space and therefore have precedence over I. C. engines for marine service.

In general, Diesel-powered ships are not as maneuverable as steamships. For this reason a larger portion of freighters and auxiliary naval craft than fighting craft are motorships.

Although there are several types of reciprocating I. C. engines which differ from each other in details so important that they are considered as different classes of machines, they all operate on the same basic principle. The air charge is rapidly compressed and consequently heated to a high temperature, the fuel is burned at high temperature, and the hot combustion products then expand against a piston and crank arrangement to produce useful work.

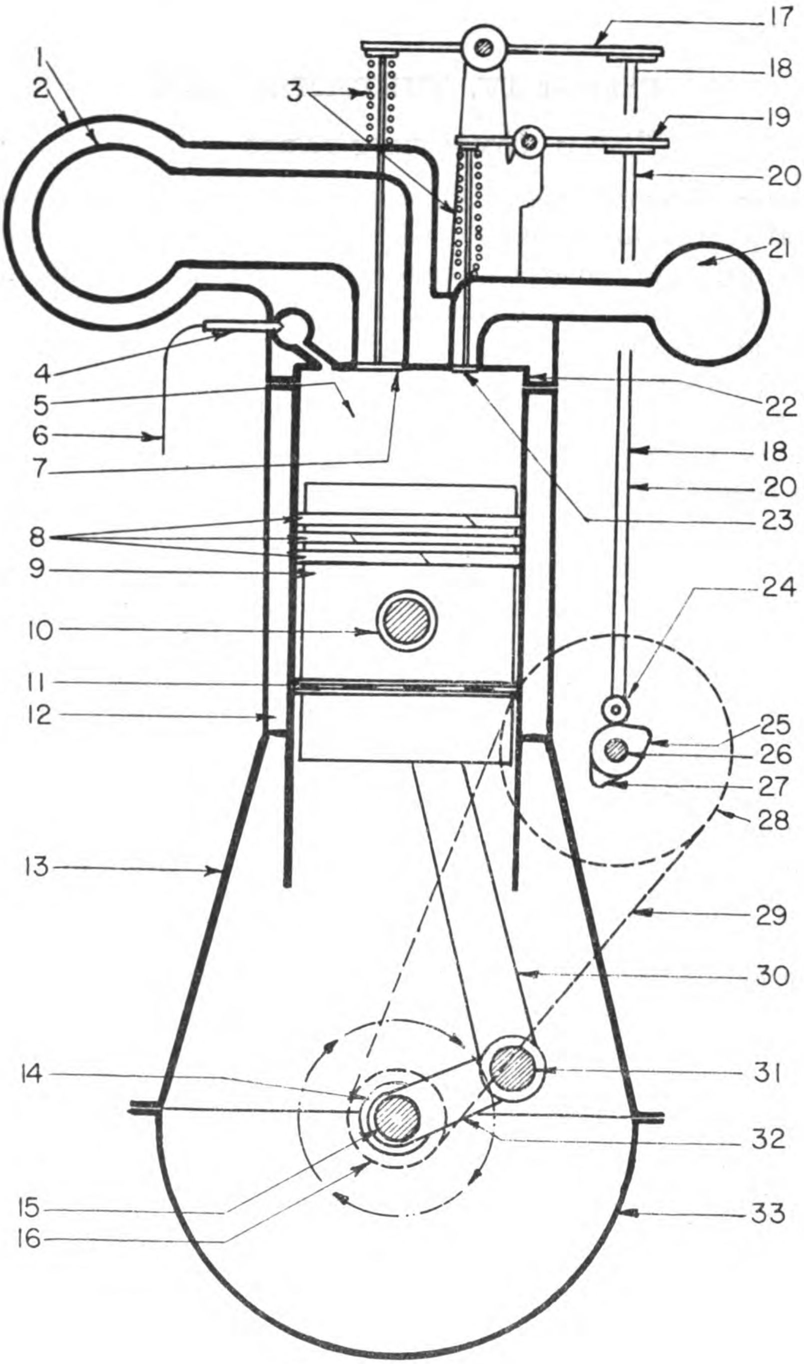

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Figure 1.

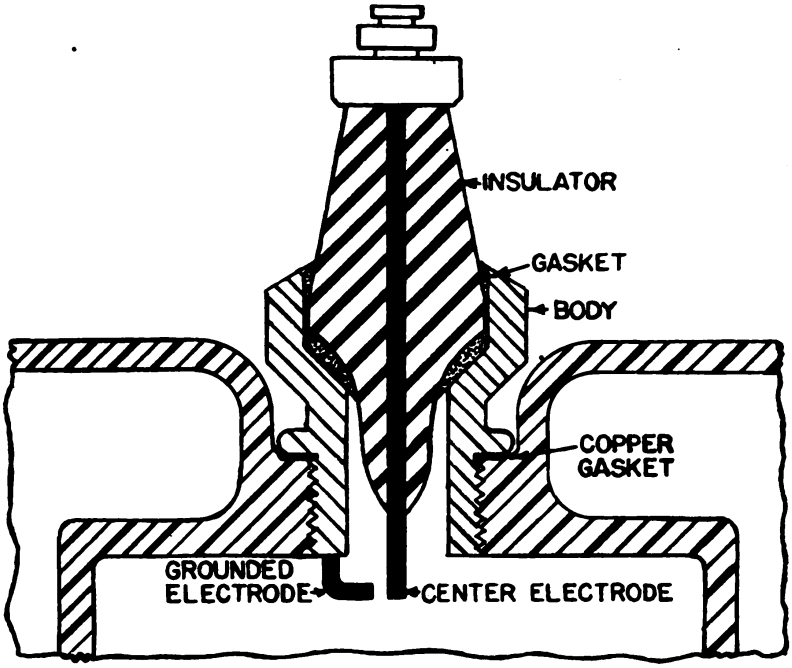
PARTS SHOWN ON FIGURE 1

1. Exhaust manifold.
2. Manifold water jacket.
3. Valve springs.
4. Fuel injection nozzle.
5. Combustion space.
6. Fuel line.
7. Exhaust valve.
8. Piston rings.
9. Piston.
10. Piston (or wrist) pin.
11. Oil ring.
12. Water jacket.
13. Engine frame.
14. Main bearing.
15. Crankshaft.
16. Timing chain sprocket.
17. Exhaust valve rocker arm.
18. Exhaust valve push rod.
19. Intake valve rocker arm.
20. Intake valve push rod.
21. Intake manifold.
22. Cylinder head.
23. Intake valve.
24. Cam follower.
25. Intake valve cam.
26. Camshaft.
27. Exhaust valve cam.
28. Timing chain sprocket.
29. Timing chain.
30. Connecting rod.
31. Crankpin.
32. Crank arm.
33. Crankcase.

IMPORTANT ENGINE PARTS

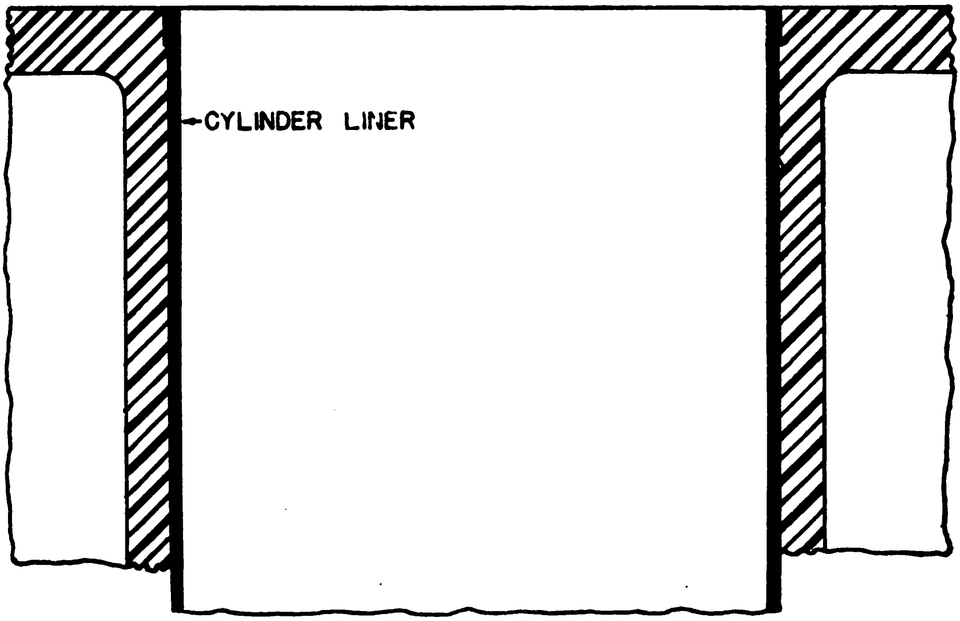
The combustible mixture of air and fuel is burned in a *cylinder* which is closed at one end by a fixed *cylinder head* and at the other by a reciprocating *piston*. A gastight seal between the piston and the cylinder is effected by *piston rings* lubricated and sealed with engine oil. Pressure is transmitted from the burning air and fuel charge via the piston to the connecting rod through the *wrist pin* (or *piston pin*) in the piston assembly on which the connecting rod is free to oscillate. The other end of the connecting rod pivots on a crank pin of the crankshaft, and by this linkage the reciprocating motion of the piston is translated into a continuous rotary motion of the crankshaft. A *flywheel* of considerable inertia is affixed to the crankshaft and acts to reduce speed fluctuations by storing kinetic energy during periods of acceleration and returning it to the shaft during periods of deceleration. A *camshaft* is driven directly from the crankshaft by *timing gears* or by a chain drive. Through *cam followers*, *tappets*, and *rocker arms*, the intake and exhaust *valves* are actuated by cams on the camshaft. Combustion air, or air and fuel mixture, is distributed to the intake valves through an *intake manifold*, and exhaust gases are collected from the exhaust valves or ports by an *exhaust manifold* or by individual pipes leading to an exhaust trunk. A *crankcase* is generally constructed to protect the crankshaft, bearings, connecting rods, and related parts, to provide a reservoir for lubricating oil, and to catch the oil escaping from the bearings of the moving parts. These parts are shown in figure 1.

Fuel and air for a gasoline engine are usually mixed in a *carburetor* and the mixture is supplied directly to the intake manifold. In a few recent aircraft engines the fuel is injected directly into the engine



SPARK PLUG

Figure 2.



CYLINDER LINER

Figure 3.

cylinders at the beginning of the compression stroke. In spark ignition engines, the air and fuel mixture is ignited at the end of the compression stroke by an electric spark passed between the electrodes of a *spark plug* in the upper part of the combustion space in the cylinder, as shown in figure 2. The spark is produced and timed with a *spark coil* and a *distributor*.

Fuel for compression ignition engines is sprayed directly into the combustion space at the end of the compression stroke by an injection system consisting of *spray nozzle* and valve and an *injection pump*.

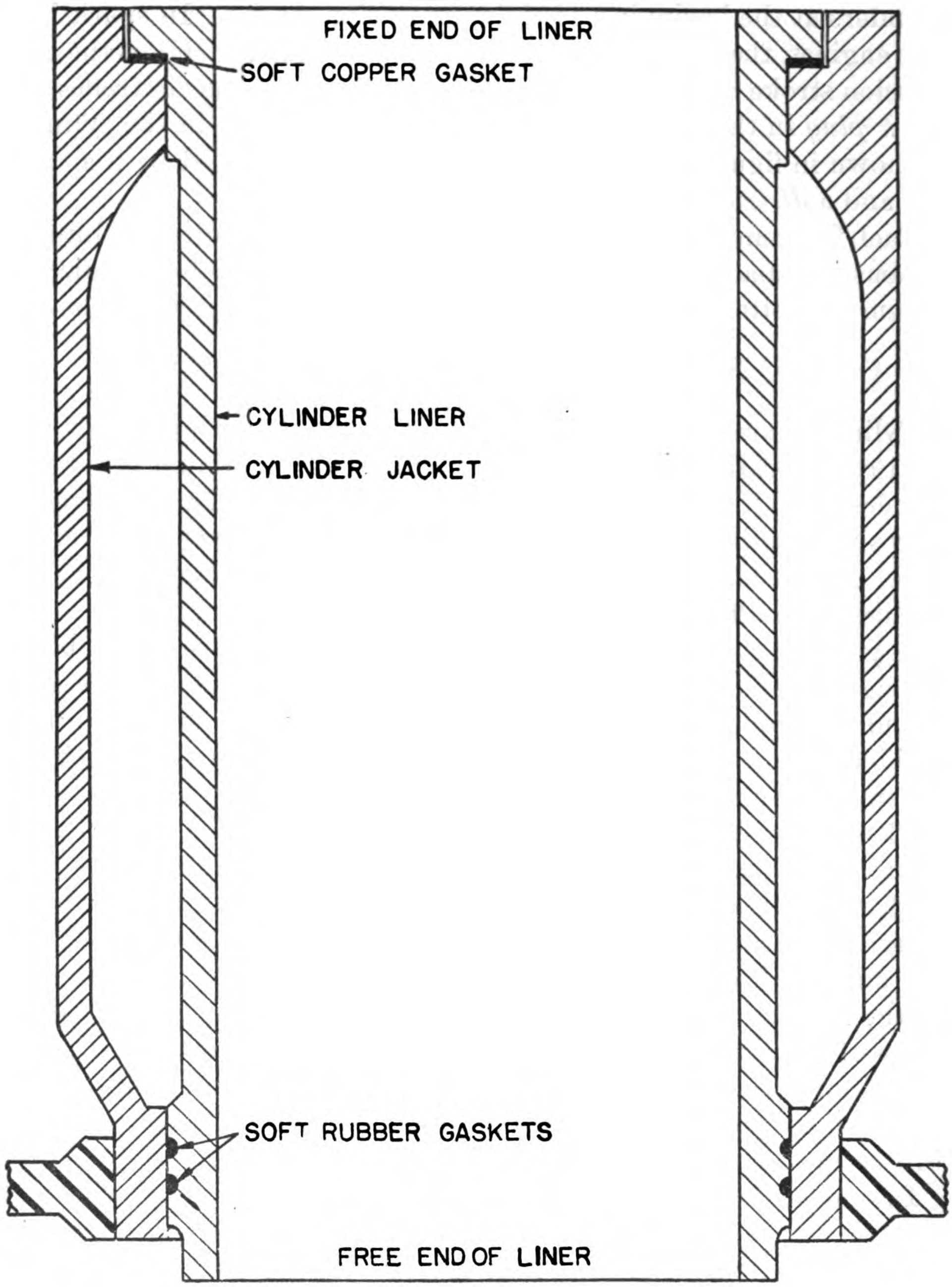
CYLINDERS

Cylinders are generally made of cast iron. If saving weight is a vital consideration, cylinders may be made of aluminum fitted with seamless steel liners, or of welded steel construction. In small engines only single-acting cylinders are used, one end being open to the crankcase. A few large slow-speed engines are double acting. Very few engines are built with less than two cylinders.

In general, cylinders for multiple-cylinder engines are cast in blocks for rigidity and simplicity of construction. Very large engines some-use unit-cylinder castings to avoid excessive weight and size of individual parts, though the possibility of carrying a spare cylinder for emergency replacement is a claimed advantage.

Close-grained cast iron has a low friction coefficient and excellent wear resistance to rubbing action. Cast-iron cylinders are usually used with liners. All cast-iron cylinders of large bore are fitted with separate liners of seamless steel tubing, fastened rigidly at the upper end and free to move longitudinally at the lower end independently of the jacket. (See figure 3.) This is an important advantage because of the great difference in temperature between the two surfaces. It is imperative that a good thermal contact be made between the liner and the jacket, as otherwise unequal heating of the cylinder liner will occur, causing scored walls and possible piston seizure.

In some makes of Diesel engines, the cylinder block acts as the principal structural member and contains the water jacket passages, but the cylinders themselves are not integral with the block. The cylinders take the form of a tube, machined outside and accurately bored and ground inside, with a flange at the upper end. The upper end is rigidly fastened in the cylinder block, and a watertight seal is obtained by a rubber gasket (or copper and asbestos gasket) clamped in a groove between the cylinder and the block. The lower end is free to move longitudinally independently of the block, and a watertight seal is obtained with a rubber gasket. This construction is illustrated in figure 4.



MOVEABLE CYLINDER LINER

Figure 4.

Pistons and cylinders are oil lubricated, and it is essential that they be kept cool enough to prevent cracking of the lubricating oil with attendant "carbon" formation. Modern compounded oils are more resistant to thermal disruption of the lubricating film than straight

mineral oils, but it is still important to maintain a wall temperature low enough to prevent rupture from this cause. For this reason all internal combustion engine cylinders are specially cooled. Two methods are employed. Air-cooled cylinders are provided with cooling fins of large surface area over which air is passed at high velocity. This construction is frequently used for aircraft engines, but rarely used for marine engines. All other internal combustion engines are liquid cooled by rapidly circulating cool liquid through cylinder jackets (and through cored passages in the cylinder head of large engines). These constructions are shown in figure 1. The cooling medium now used for aircraft engines is ethylene glycol, and for marine engines, water. A few installations now in use provide for pumping sea water through the jackets. The modern trend is to use distilled water in the jackets and to cool this in a surface type cooler with sea water. This avoids salt and mud deposits in the engine jackets, and reduces corrosion problems. The surface coolers are similar in construction to marine steam condensers, and cleaning is relatively easy.

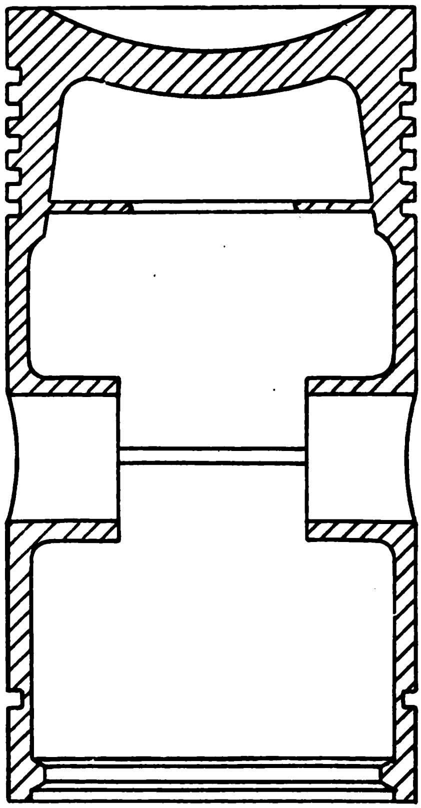
In cast-iron cylinder blocks, the jacket passages are formed by coring and they are cast integral with the block. In welded construction, they are obtained by rolled, drawn, or built-up jackets electrically welded to steel cylinder liners.

PISTONS

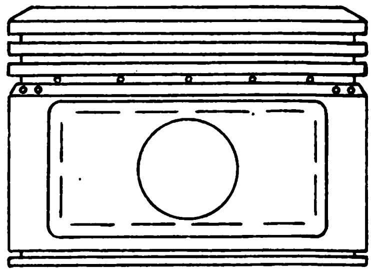
Pistons are constructed of cast iron for low-speed engines, and of aluminum alloy for nearly all high-speed engines. The piston must be designed to withstand the gas pressure developed above it during the compression stroke and the subsequent combustion of the fuel. It must be light enough to keep the inertia loads on the wrist pin and the crank pin bearings to a minimum. Ample means must be provided to conduct heat from the piston head to prevent destruction of the metal. An aircraft engine piston and a Diesel piston are shown in figure 5.

Due to the angularity of the connecting rod with respect to the axis of the cylinder as it follows the motion of the crank, there is a side thrust developed. Some means of resisting this thrust must be provided. In most single acting engines trunk pistons are used, and the side thrust is transmitted by the piston to the cylinder wall. (A trunk piston is one whose length is greater than its diameter.) Such pistons must be designed to withstand this force and to resist the resultant wear. The magnitude of the side thrust is proportional to the gas pressures developed and to the inertia forces (and hence to the weight of the moving parts and the speed of the engine). It is decreased as the ratio of connecting rod length to stroke is increased. As the connecting rod is lengthened side thrust is decreased greatly. Pistons, particularly if made of aluminum, are frequently fitted with

a piston ring at the bottom of the skirt. The component of the side thrust at the lower end of the piston is then taken by this ring instead of by the skirt itself.

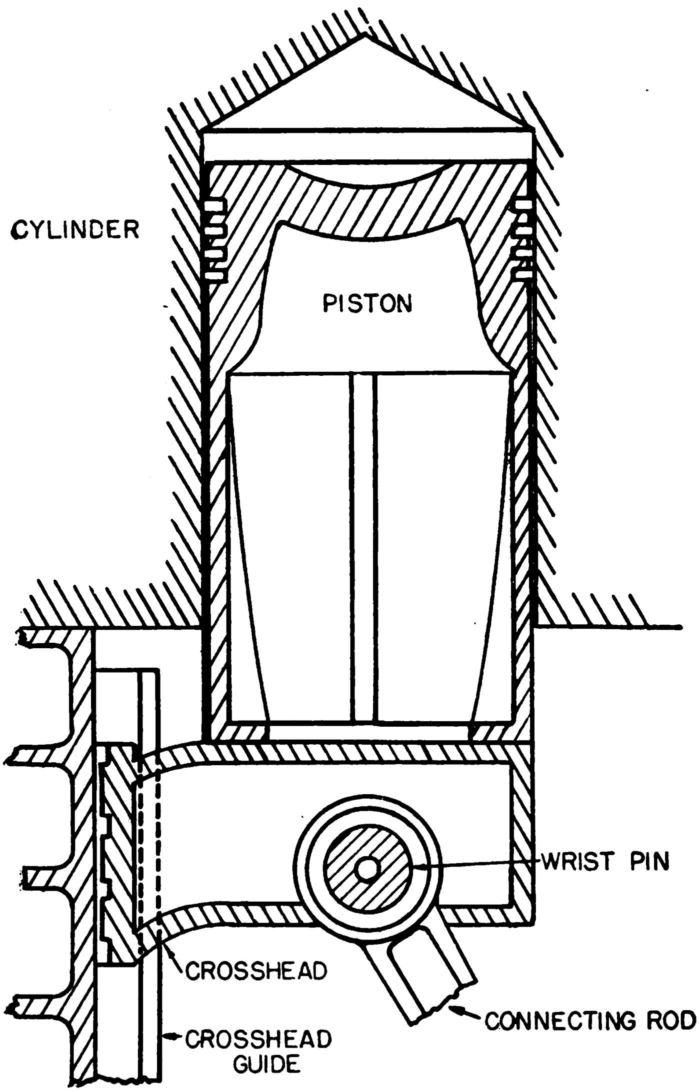


SECTION THROUGH TRUNK PISTON
FOR DIESEL ENGINE



AIRCRAFT ENGINE PISTON

Figure 5



CROSSHEAD AND PISTON

Figure 6.

An alternative construction, used by some designers of large single acting engines and on all double acting engines, is to use a separate crosshead and guides. In this case, a piston rod connects the piston to the cross head and the latter carries the wrist pin from which the connecting rod is swung. This is shown diagrammatically in figure 6.

Pistons are machined to fit the cylinder diameter with clearance enough to compensate for the expansion due to heat and to permit lubrication between it and the cylinder walls. Aluminum pistons, with their high coefficient of thermal expansion must be loosely fitted,

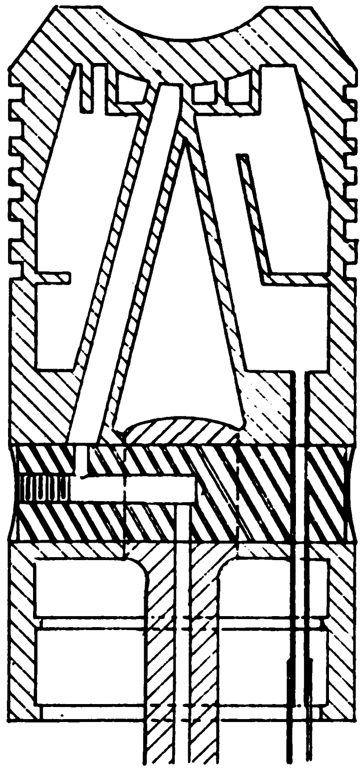


Figure 7

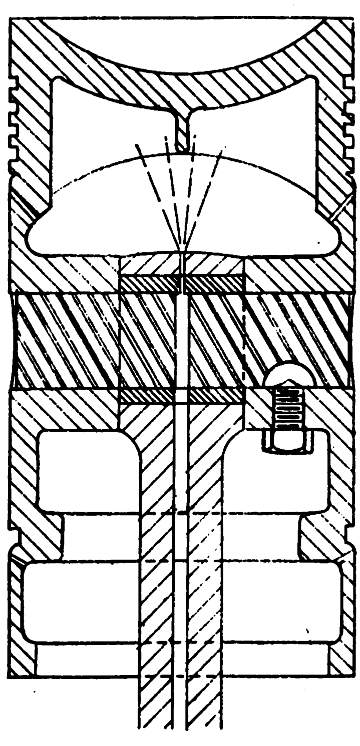


Figure 8

and there is therefore a tendency toward piston slap. This is sometimes controlled by splitting the skirt of the piston. The fault is prevented in the autothermic piston by using low-carbon steel strips in combination with aluminum alloy to control skirt expansion.

Most pistons are made with a slight taper, being larger on the crank end than on the head end where the highest temperatures are encountered.

The temperatures at the center of the piston head are the highest encountered in an internal-combustion engine except in the burning gases themselves. It is essential that adequate means be provided for conducting this heat from the piston to avoid deterioration of the metal. In small- and medium-size engines, heat transfer through the piston head to the rings and thence to the cylinder walls is mainly relied upon. This is supplemented by air cooling on the under side of the piston due to the pumping action as it reciprocates, and by cooling by the oil escaping from the wrist pin bearing and splashed by the cranks spinning in the crankcase.

In large engines some form of positive cooling is required. One method is to core the piston head and supply a flow of oil to this core from the wrist pin oil supply line in the connecting rod, a construction shown in figure 7. Another method is to supply excess oil through the forced feed supply to the wrist pin and to spray jets of this oil on the lower side of the piston. See figure 8. Some large slow speed engines on shore use water cooled pistons, the connections to the piston head core being made through telescoping metal pipes. This method has been used on ships, but is in disfavor because of the tendency of salt water and oil to form emulsions if leaks occur.

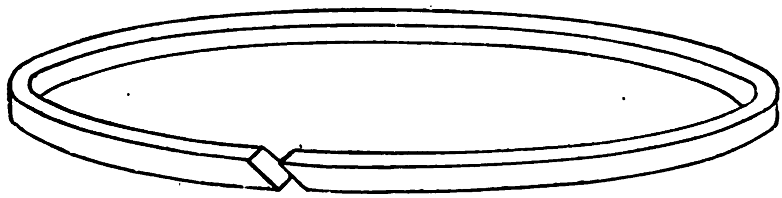
PISTON RINGS

Since it is essential to fit the pistons of an internal-combustion engine in the cylinders with an ample clearance to provide for thermal expansion, a special means of forming a gastight seal between the pistons and cylinders is required. This seal is effected by oil lubricated piston rings. The rings fit in grooves in the piston and are pressed outward against the cylinder partly by their own springiness and partly as the result of the gas pressure which builds up behind them. The rings, though they fit the cylinder closely, do not in themselves provide an adequate gastight seal. This is effected by the engine oil that lubricates the cylinder walls and rings and forms a seal by flooding the spaces between rings where it is held by capillary action.

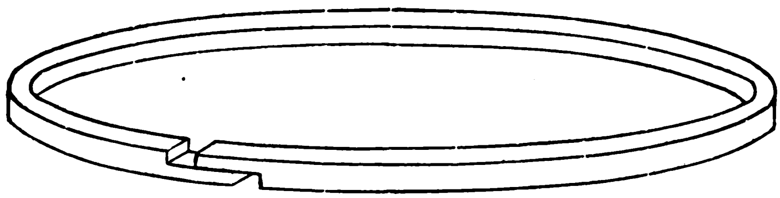
The number of types of piston rings is legion, and superlative advantages are claimed for each type. Some are considerably better

than the usual snap ring, but plain cast iron snap rings are almost universally used.

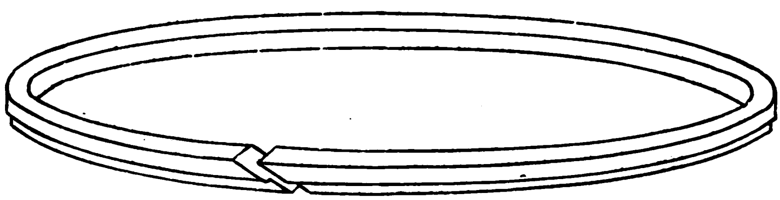
The cylinder walls of most high and medium speed engines with a bore less than 12 inches are lubricated by oil splashing from the ends



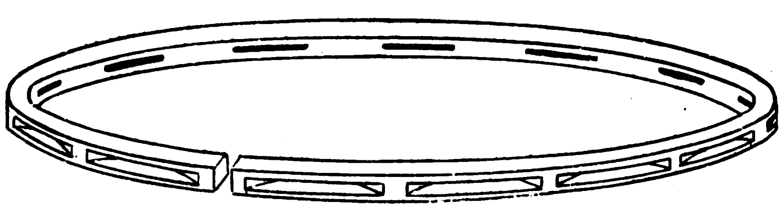
DIAGONALLY-CUT COMPRESSION RING



LAP-JOINT COMPRESSION RING



OIL RING



SLOTTED OIL RING

PISTON RINGS

Figure 9.

of the crankpin bearings and from the crankcase. There is no way to control the amount of this lubrication, and it is consequently quite possible that cylinder may receive an excessive oil supply. This oil

gradually works up into the combustion space where it is cracked and produces "carbon." Oil-control rings, if properly fitted, will scrape off the excess oil and will help to distribute oil uniformly around the cylinder wall.

A few types of rings are shown in the accompanying sketch, figure 9.

Three or four rings are usually fitted on each piston of a gasoline engine, and on Diesel engines as many as six or seven may be used. The upper rings are usually plain snap rings for gas seals, and the ring just above the wrist pin (and one at the bottom of the skirt if fitted) are oil-control rings.

The grooves in which the oil rings are installed are provided with drain holes leading to the inside of the piston skirt so that excess oil may drain off and be returned to the crankcase.

Piston-ring leakage also has a very bad effect on the parts of the engine. The hot gas passing the rings tends to destroy the lubricating oil film on the cylinder, and causes the rings and piston to become unduly hot from the increased friction resulting from faulty lubrication. The hot gas flowing past the piston also causes unsatisfactory working conditions for the wrist-pin bearing, and since the exhaust gases contain a great deal of water vapor, there is a tendency toward accumulation of water in the lubricating oil unless crankcase temperatures are high enough to prevent it. These causes may lead to scoring and possible seizure of the piston with resulting serious damage to the engine. Proper fitting of the piston rings is therefore of vital importance.

If the piston rings have sharp edges, they act to scrape too much oil from the cylinder walls. It is therefore advisable to round the outside edges of the rings very slightly to prevent this action. A radius of one or two hundredths of an inch is sufficient.

WRIST PINS

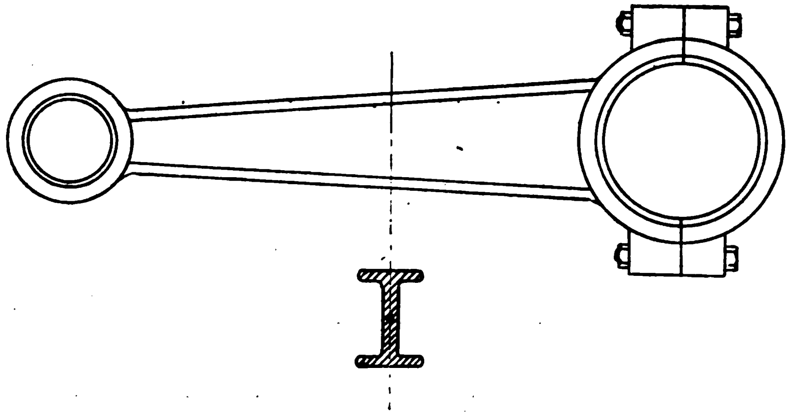
The wrist pin is used to connect the connecting rod to the piston and it forms the pivot around which the connecting rod oscillates as it follows the motion of the crank. The pin is usually made of alloy steel, machined, hardened, and precision ground and lapped. In various designs it is rigidly fastened to the piston, clamped in the end of the connecting rod, or free to rotate in either member. In any case it must be secure against endwise motion, for if one of its ends should come in contact with the piston wall, it would soon cut and score the wall.

Wrist-pin bearings in small engines are usually bronze-lined oil-lubricated cylindrical bearings. In some recent designs of large high-compression engines, needle bearings are used. Needle bearings are a type of antifriction bearing with a very large number of small

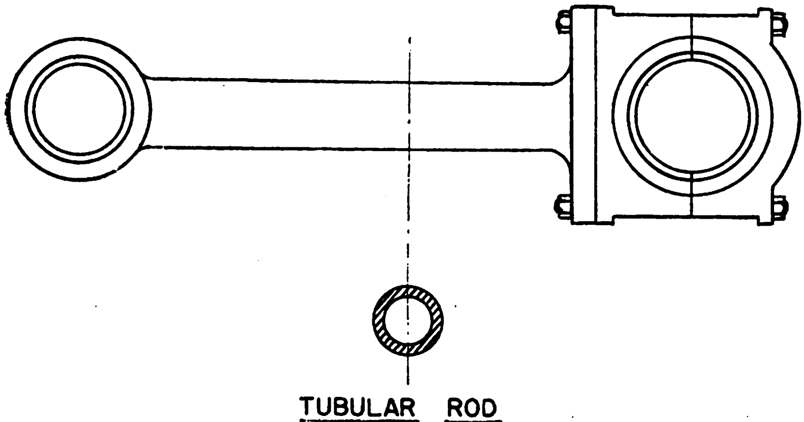
diameter rollers. They are very effective for bearings in which oscillatory motion is encountered.

CONNECTING RODS

In small engines the connecting rod is an alloy steel forging. It serves as the connecting link between the piston and the crankshaft. It must be designed to resist compression loads resulting from the



H-SECTION ROD

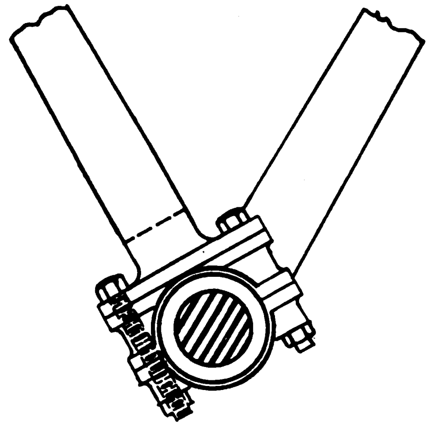


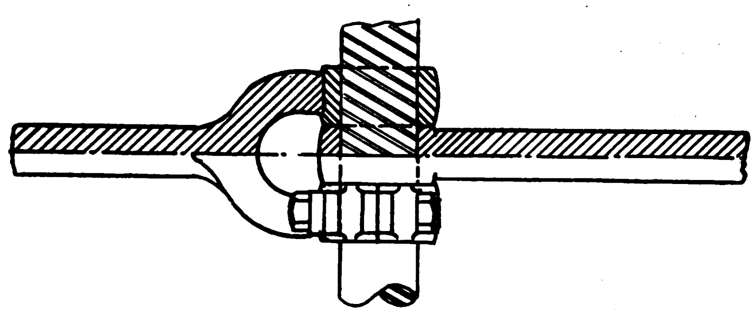
TUBULAR ROD

CONNECTING RODS

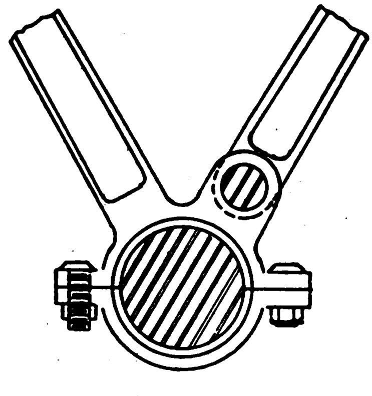
Figure 10.

CONNECTING RODS





FORKED TYPE



MASTER ROD TYPE

Figure 11

gas pressure acting on the piston, and must also withstand tension load resulting from deceleration of the piston at the top of its stroke. At very high speeds the inertia forces become the critical forces acting on the reciprocating parts of the engine, and failure in tension sometimes results when an engine races. In high-speed engines the rod is made either tubular or with an H section. This is illustrated by figure 10.

Most large engine connecting rods are drilled to provide a passage for oil from the crankpin bearing to the wrist pin bearing.

In Vee, W, X, H, and radial engines, two or more connecting rods are connected to the same crankpin. There are three methods of connecting more than one connecting rod to a single crankpin. The first two methods are applicable only to two rods, the third to any number required.

(1) Two connecting rod ends may be located on the same crankpin side by side. This necessitates a slight offsetting of the two banks of cylinders.

(2) One rod may be forked and the two halves of its bearing straddle the single bearing of the mating rod.

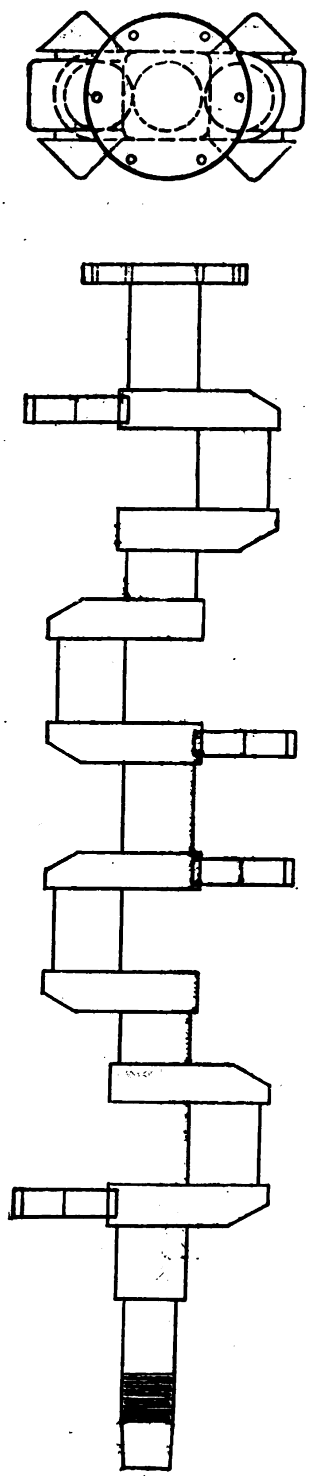
(3) One master rod may have a bearing on the crankpin and be provided with one or more lugs to which shorter rods are pivoted for the other cylinders. These types are shown in figure 11.

CRANKSHAFT

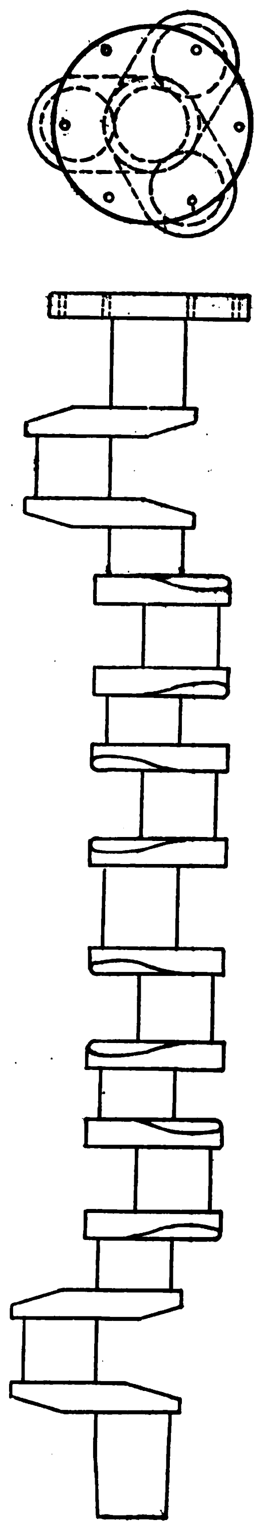
The crankshaft is used to translate the reciprocating motion of the pistons to a rotary motion of the output shaft. It must be very carefully designed and constructed. Either excessive limberness or excessive rigidity may lead to failure. Two crankshafts are shown in figure 12.

Crankshafts are generally made of forged alloy steel, although some recent small engines use cast-iron crankshafts successfully. They are often made hollow, especially in large engines, to reduce weight, insure homogeneity of material, and to provide an oil passage. Crankshafts for engines up to 1,000 horsepower are usually made from integral forgings or castings. Built-up crankshafts are sometimes used on very large engines.

Nearly all crankshafts are now balanced both statically and dynamically. Counterweights may be used for crankshafts having fewer than six crankpins. Due to the natural distribution of the masses of the component parts, six- and eight-throw chankshafts can be perfectly balanced without counterweights. Both static and dynamic balancing are required in order to secure a wide range of satisfactory operating speeds.



COUNTERBALANCED CRANKSHAFT FOR FOUR CYLINDER ENGINE



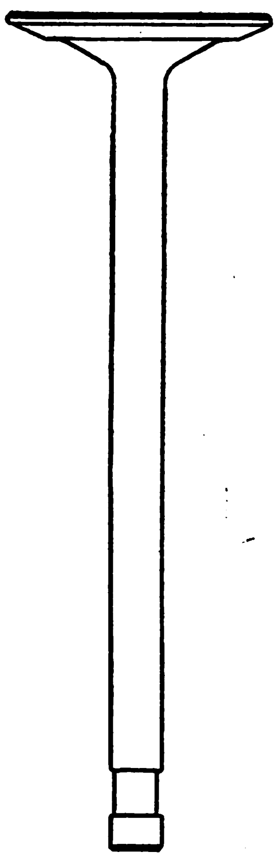
CRANKSHAFT FOR SIX CYLINDER ENGINE

Figure 12

All multiple cylinder engines have critical speeds at which harmonic oscillations develop. Each impulse, being in phase with the harmonic oscillation, increases the amplitude of the oscillation. This produces excessive torsional flexure of the shaft, and if the engine is operated at one of these speeds for any considerable length of time, failure is almost certain to occur. Most large engines have plaques attached which tabulate the speeds at which the engine must not be run. It is advisable to avoid these speeds by about 10 r. p. m. or more if possible. The engine may be accelerated, passing through a critical speed without harm if the acceleration is reasonably rapid.

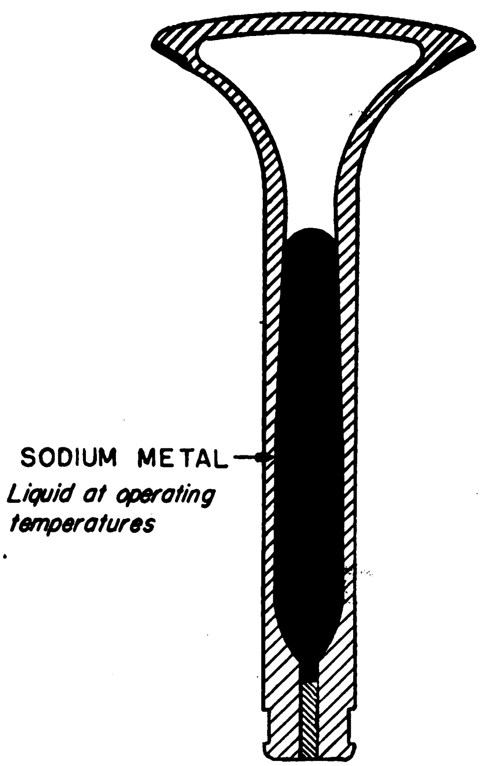
VALVES

Valves are provided to admit the mixture to the cylinder at the proper time and to allow exhaust of the combustion products at the end of the power stroke. The valves of modern internal combustion



INTAKE VALVE

Figure 13.



SODIUM COOLED EXHAUST VALVE

Figure 14.

engines are of the poppet type, and are made of alloy steels, such as silichrome, tungsten steel, and chrome-vanadium steel. They must resist corrosion and deterioration and retain their hardness at high temperatures, and should not warp. A poppet intake valve is shown in figure 13.

Exhaust valves for some large engines are cored and are cooled by circulating oil or water through them. Recently, valves for aircraft engines have been made with a hollow core in the forging which is partially filled with metallic sodium. The sodium is liquid at operating temperatures and provides for very effective heat transfer to the water cooled parts of the cored cylinder head casting in which the valve stems slide. This construction is shown in figure 14.

2. TYPES OF INTERNAL COMBUSTION ENGINES

CLASSIFICATION BY METHOD OF IGNITION

Internal-combustion engines may be divided into two main classes by the means employed to ignite the fuel at the start of combustion.

Spark ignition engines.—These are the gasoline engines. Two features are peculiar to this class; (1) the fuel is mixed with the combustion air prior to compression, and (2) combustion is initiated by an electric spark timed so that ignition occurs near the end of the compression stroke.

Compression ignition engines.—These are the oil engines. In these engines the combustion air is compressed to a high temperature and pressure, and the fuel is subsequently injected as a fine spray at or near the end of the compression stroke. In the case of a Diesel engine, the compression ratio is high enough to heat the compressed air to a temperature that ignites the fuel directly. In the case of the hot-bulb (or semi-Diesel) engine, the compression temperature is not high enough to ignite the fuel, but the fuel is sprayed against a piece of metal which is sufficiently insulated from the cool parts of the engine to be red hot from the previous operation of the engine.

CLASSIFICATION BY NUMBER OF PISTON STROKES PER CYCLE

There are two engine classifications based on the number of piston strokes per cycle.

Four-cycle.—Four-cycle engines are those requiring four piston strokes per complete cycle. All gasoline engines (except for lawn mowers, outboard boat motors, and similar applications) are four-cycle engines. The majority of compression-ignition engines in naval service operate on a two-stroke cycle.

The sequence of events for four-stroke cycle engines is as follows:

Gasoline engine	*Diesel engine*
1. On the first down stroke of the piston, the cylinder is filled with a mixture of volatile fuel and air.	1. On the first down stroke of the piston the cylinder is filled with fresh air.
2. On the up stroke the combustible mixture is compressed to a moderate pressure. Near the top of the stroke the mixture is ignited by the passage of an electric spark. Combustion occurs while the piston is at the top of its stroke.	2. On the up stroke the air is compressed to a high pressure. Near the top of the stroke the injection of fuel is initiated. The fuel ignites spontaneously due to the heat of compression.
3. On the following down stroke the gases are allowed to expand, doing work on the piston. Near the end of the stroke the exhaust valve is opened, allowing the pressure in the cylinder to drop nearly to atmospheric.	3. On the following down stroke combustion continues until the injection is stopped. The gases are then allowed to expand. Work is done on the piston during all of the stroke. Near the end of the stroke the exhaust valve is opened, allowing the pressure in the cylinder to drop.
4. On the succeeding up stroke the burned gases are forced out of the cylinder.	4. On the succeeding up stroke the burned gases are forced out of the cylinder.

These events are shown by the diagrams of figures 1 and 2.

Two-cycle.—Two-cycle engines, those requiring two piston strokes per complete cycle, dominate the large-engine field (over 1,000 horsepower).

The sequence of events for two-stroke-cycle Diesel engines is as follows:

> 1. On the up stroke air is compressed to a high pressure. Near the top of the stroke the injection of fuel is initiated. The fuel ignites spontaneously due to the heat of compression.
>
> 2. On the down stroke combustion continues until the injection is stopped. The gases are then allowed to expand. Work is done on the piston during all of the stroke. Near the end of the stroke the exhaust valve is opened, allowing the pressure in the cylinder to drop nearly to atmospheric. The scavenging valve is then opened and the exhaust gases swept out of the cylinder. The exhaust valve is closed and the cylinder fills with air to the pressure in the scavenging line. The scavenging valve is closed as the piston begins to move rapidly on the compression stroke.

Some sort of low-pressure air supply is required for scavenging a two-cycle engine. This may be obtained by an auxiliary rotary or reciprocating blower, which may or may not be directly driven by the main engine, or it may be obtained by crankcase compression.

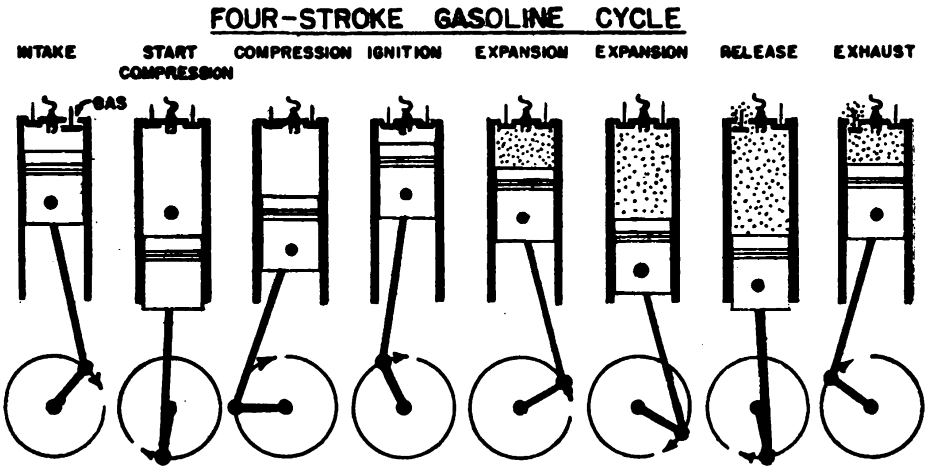


Figure 1.

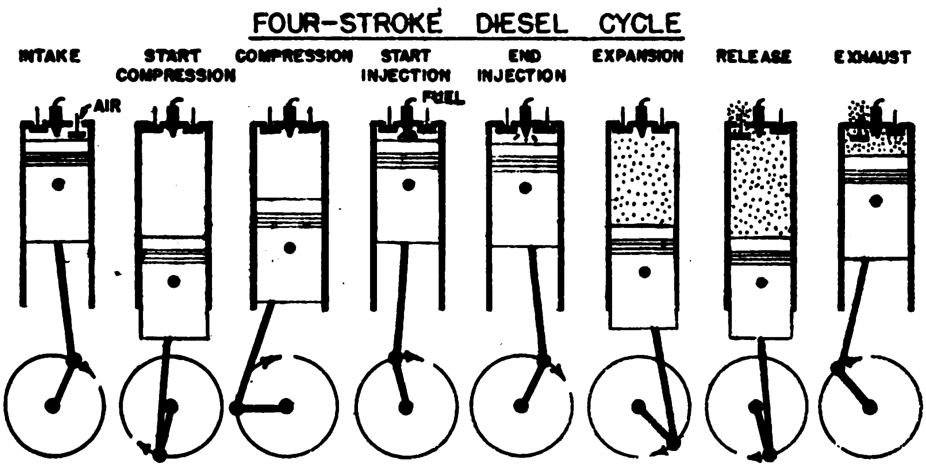


Figure 2.

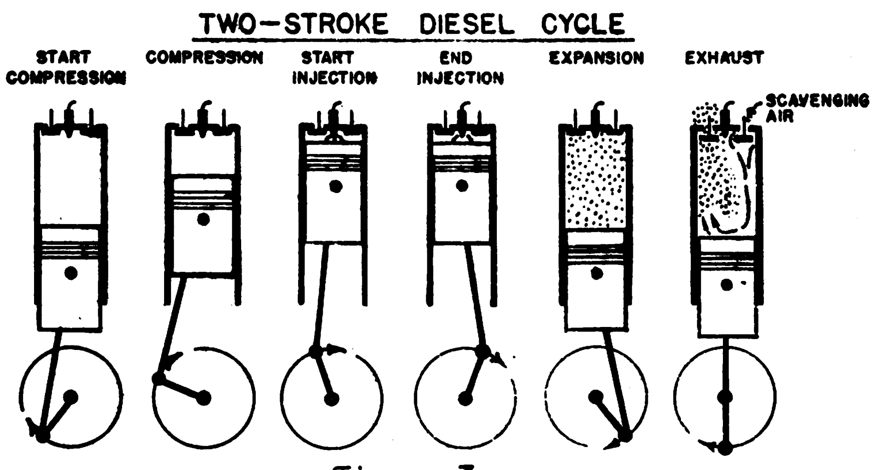


Figure 3.

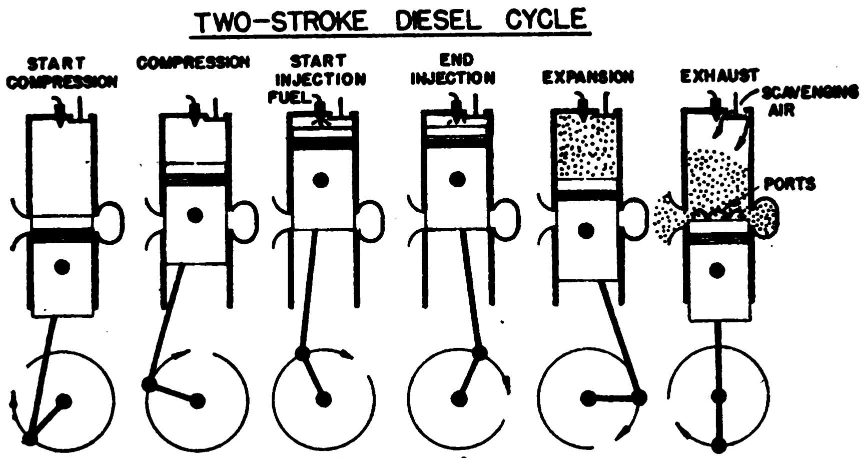


Figure 4.

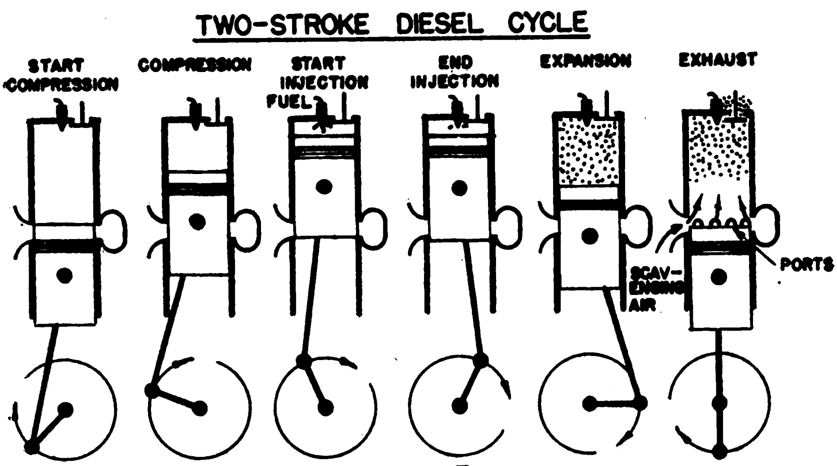


Figure 5.

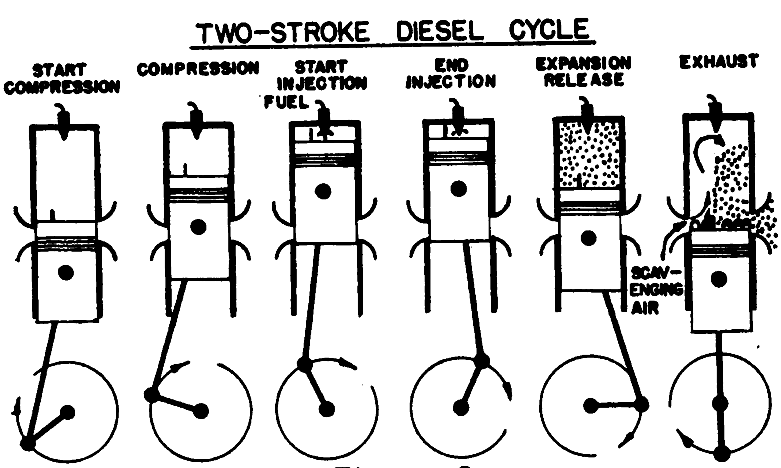


Figure 6.

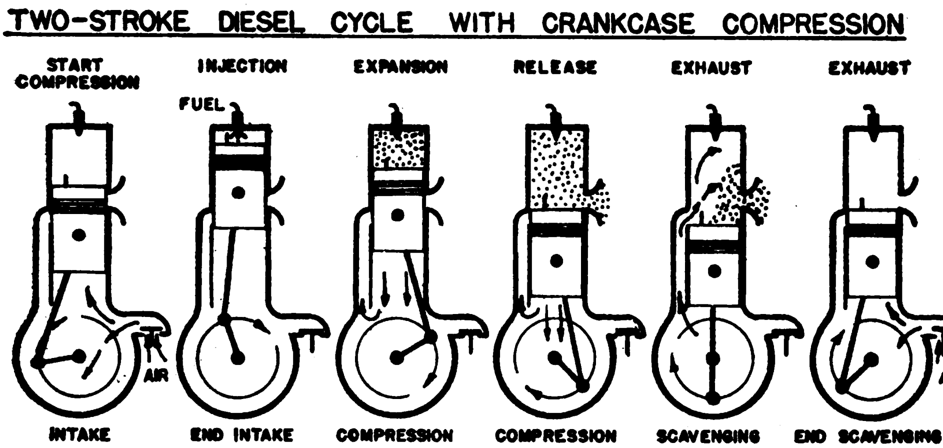


Figure 7.

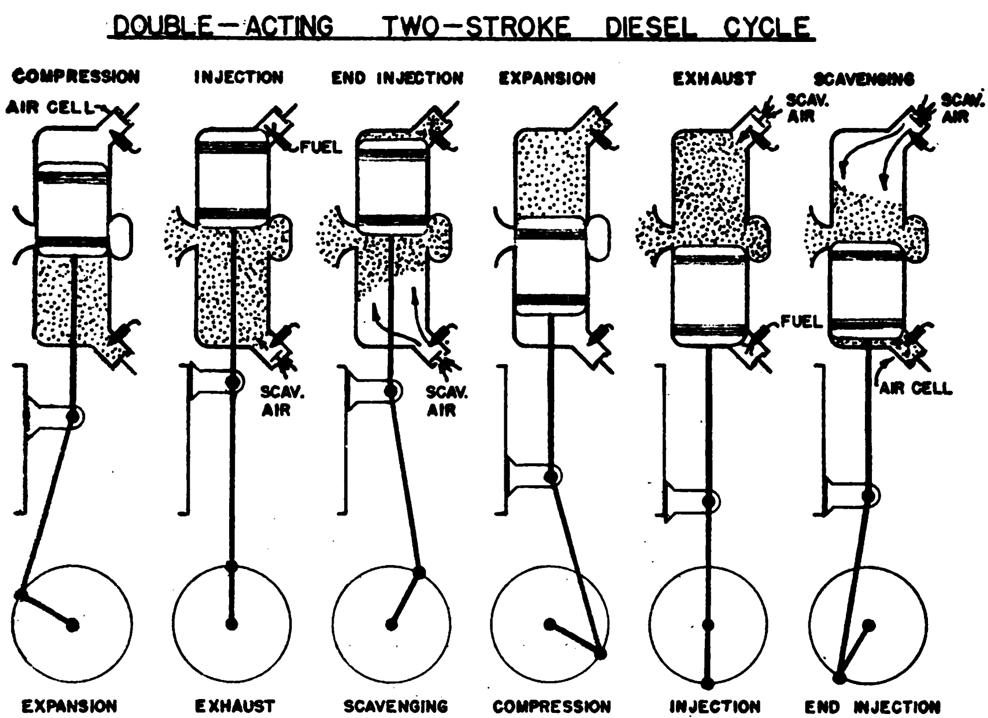


Figure 8.

There are several arrangements of scavenging and exhaust valves for two-cycle engines. These are: (1) scavenging and exhaust valves located in the cylinder head (fig. 3); (2) scavenging valve in the head, and exhaust ports in the cylinder (fig. 4); (3) exhaust valve in the head, and scavenging ports in the cylinder; and (4) both exhaust and scavenging ports in the cylinder without auxiliary valves (fig. 6). Crankcase compression engines use the fourth valve arrangement (fig. 7).

The sequence of events for the various arrangements of two-cycle Diesel engines are shown by the diagrams.

CLASSIFICATION BY PISTON ACTION

Single-acting engines.—Single-acting engines use only one end of the cylinder and one face of the piston for the development of power. The engines in figures 1–7 are all single acting. Nearly all internal combustion engines are single acting.

Double-acting engines.—Double-acting engines use both ends of the cylinder and both faces of the piston for the development of power. Figure 8 is a diagram of a double-acting two-cycle engine. Because of the serious mechanical difficulties of maintaining a lubricated piston rod and packing gland operating in the combustion space of an engine, double-acting engines are usually built in large and slow-acting designs.

Opposed piston engines.—Recently, many engines having two pistons per cylinder, driving two crankshafts (usually located on top and bottom of cylinder block) have been installed in naval service. This design approximates the performance of a corresponding double-acting engine, and eliminates trouble at the piston rod packing gland (see fig. 10).

CLASSIFICATION BY COOLING MEDIUM

Liquid-cooled engines.—Liquid-cooled engines are cooled by a liquid coolant circulated through passages in the cylinder block and head which are provided for the purpose. The liquid is in turn cooled in a heat exchanger in which the heat is transferred to air or to sea water. The liquid coolants which are most used are water and ethylene glycol.

Air-cooled engines.—Air-cooled engines are provided with fins which give a large surface area to the portions of the engine which are to be cooled. Air is circulated over the fins at high velocity. Air-cooled engines are not widely used for marine service.

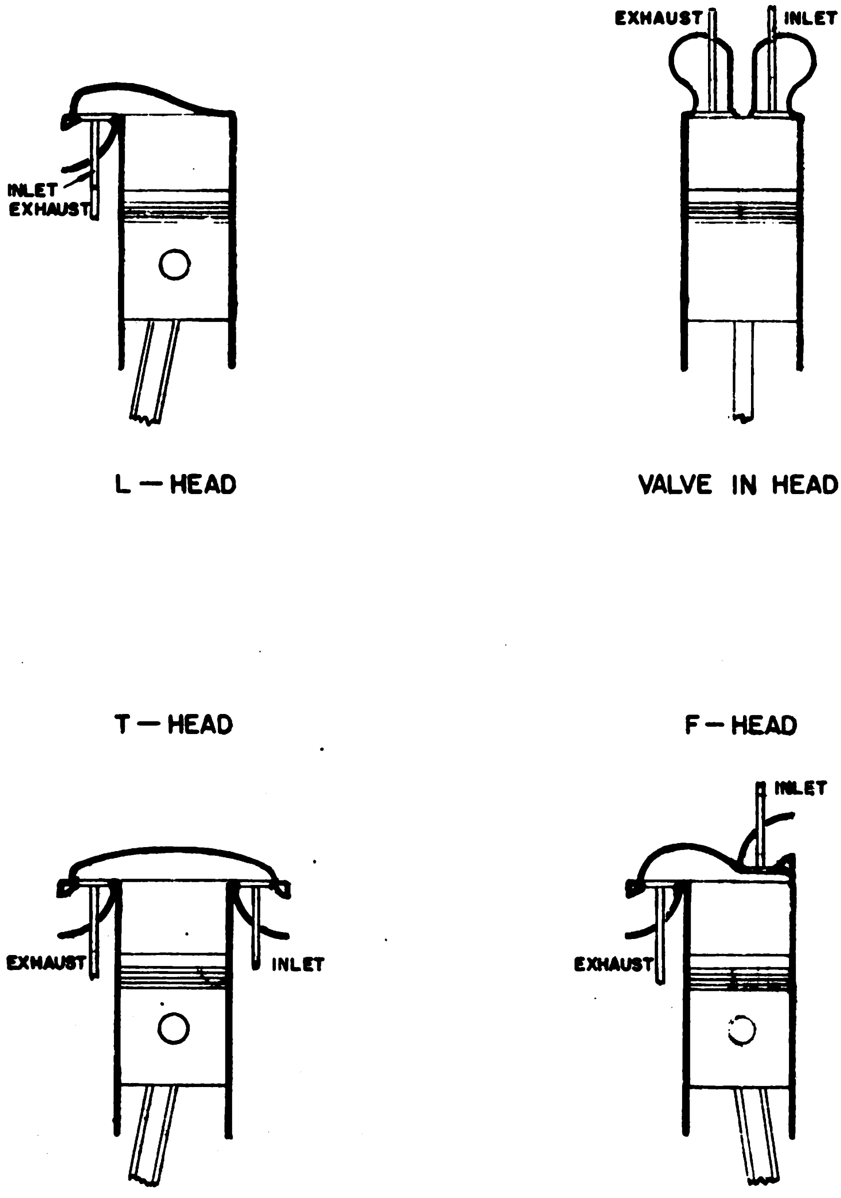
CLASSIFICATION BY VALVE ARRANGEMENTS

In addition to sleeve valve engines, which are not plentiful, there are four recognized types of valve arrangements. These are shown diagrammatically in figure 9.

L-head.—The majority of small engines are L-head engines. Only one camshaft is required, the valves being lifted directly from their seats by a simple cam follower. Exhaust and intake manifolds are crowded together.

Valve-in-head.—The majority of large engines are valve-in-head engines, and many small engines also use this construction. Only one camshaft is required, the valves being operated through rocker arms. This design offers the best combustion chamber arrangement, and there

is less resistance to entry of the charge and exhaust of waste gases than in other engines. Exhaust and intake manifolds are complicated.



VALVE ARRANGEMENTS

Figure 9.

T-*head*.—Very few engines are made with this construction. Two camshafts are required. The manifolds are simple, and the engine parts are easily reached for servicing, but the combustion chamber shape is poor.

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F-*head.*—A few engines use this construction. One camshaft is required, the inlet valve being operated through a rocker arm. In some designs the incoming gases pass directly over the exhaust valve, aiding in cooling it and in vaporizing any liquid fuel droplets present in the entering charge. Large valves may be used.

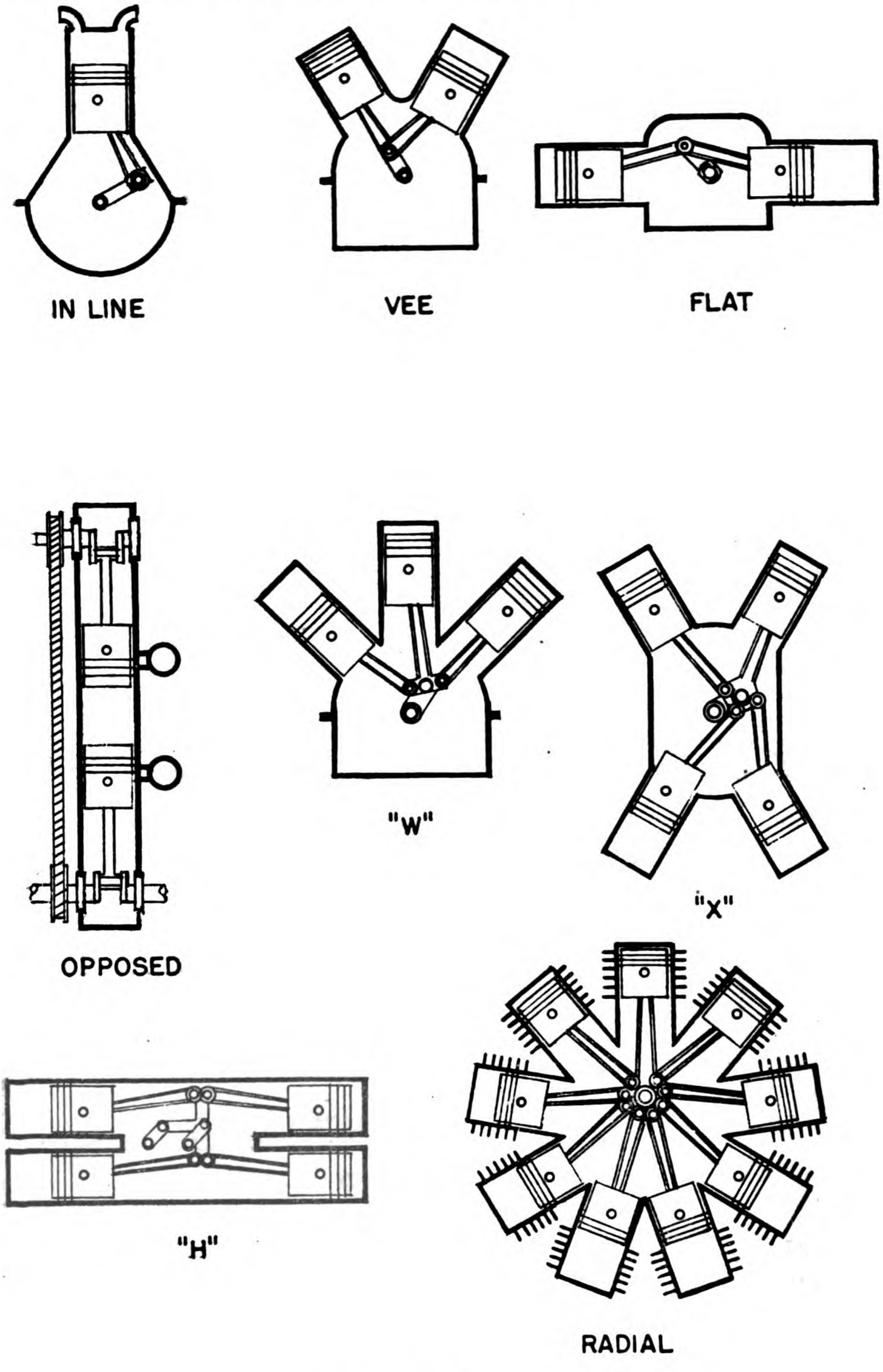


Figure 10

CLASSIFICATION BY CYLINDER ARRANGEMENT

It is not practical to construct engines with more than eight cylinders in line, because of the difficulty of making a sufficiently rigid frame and because of excessive limberness of the crankshaft. Various arrangements are therefore used with two or more connecting rods attached to each crankpin. The various arrangements are shown in figure 10.

Cylinders in line.—The construction normally used for engines with six cylinders or less. Eight-cylinder engines are about equally divided between in line and vee arrangement.

Vee.—The most common arrangement for engines with 8 to 16 cylinders. Two- and four-cylinder engines are sometimes built with the vee design. The angle between banks may vary from 30° to 180°.

Flat.—A vee enginee with 180° between banks. Now used for truck and bus engines, and for aircraft engines.

Opposed piston.—An arrangement with two single-acting pistons operating opposed to each other in the same cylinder. Constructions using one or two crankshafts are used. This is an unusual design used for aircraft and marine Diesel engines.

W.—An arrangement with three banks of cylinders grouped fanwise around a single crankshaft. This design is used for aircraft engines.

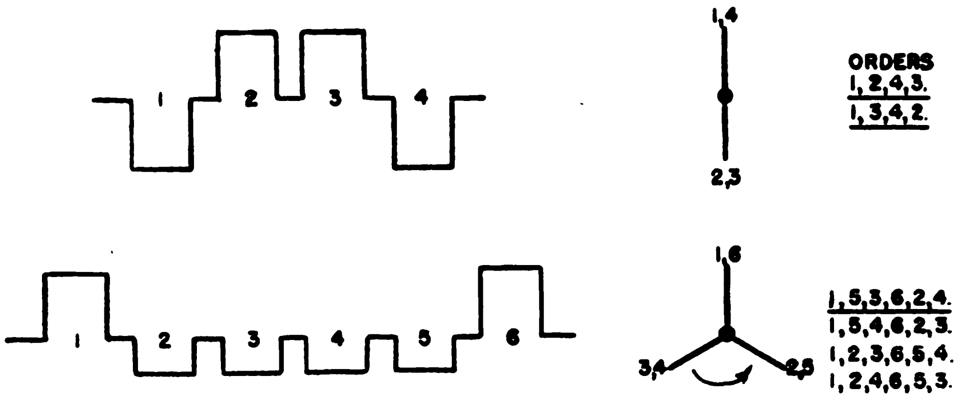
X.—An arrangement with four banks of cylinders grouped around a single crankshaft. This is an aircraft engine design.

H.—Four banks of cylinders arranged for tandem 180° operation with two crankshafts, or for operation with one master crankshaft and a stabilizing crankshaft. These are aircraft engine designs.

Radial.—An arrangement with one or two rows of an odd number of cylinders arranged around single crankpins. This is primarily an aircraft engine design for air-cooled operation.

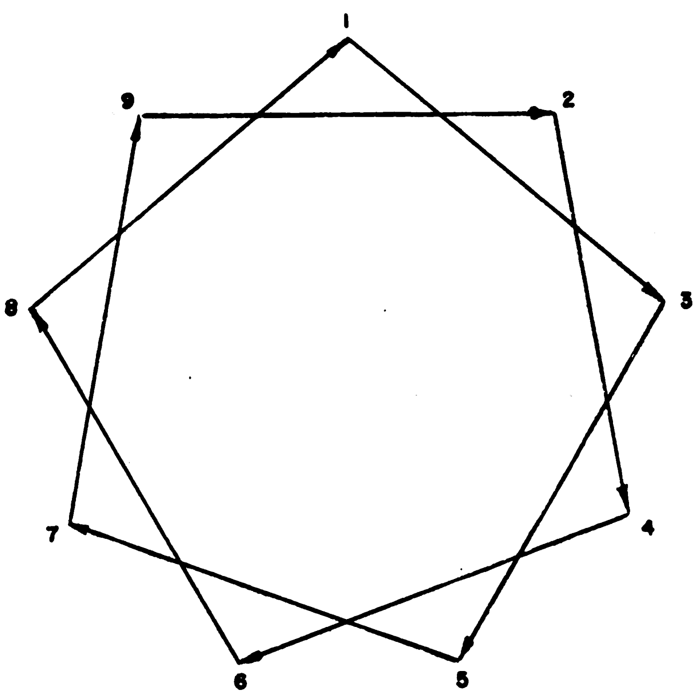
FIRING ORDER OF CYLINDERS

The number of possible firing orders for any engine is dependent upon the number of cylinders in the engine and the arrangement of the cranks. To secure uniform rotation in multicylinder engines, the power impulses should be equally spaced in respect to the angle of crankshaft rotation. Further it is desirable to avoid the occurrence of successive explosions in adjacent cylinders in order to minimize interference with exhaust gases passing through the manifold, and to produce the minimum torsional oscillation of the crankshaft.



FIRING ORDERS FOR IN LINE ENGINES

Figure 11.



FIRING ORDER FOR RADIAL ENGINE

Figure 12.

Two common forms of crankshafts for in-line engines, the possible firing orders, and the preferred firing order for each are shown in figure 11.

The firing order for a radial engine is shown in figure 12. The reason why this engine can have only an odd number of cylinders will be made clear by an attempt to draw the corresponding diagram for an even number of cylinders.

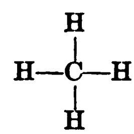
Chapter V. PROPERTIES OF PETROLEUM

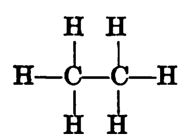
CHEMISTRY

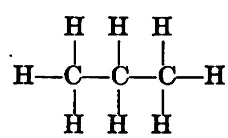
Petroleum is composed mainly of compounds of carbon and hydrogen known as hydrocarbons. These can be grouped into certain classes or series, differing within the series in the number of carbon and hydrogen atoms in a moleculte. Within a given series, increase in molecular weight is accompanied by a rise in the melting and boiling points, by an increase in viscosity, and by a decrease in volatility. Hydrocarbons in gasoline have from 4 to 14 or more carbon atoms per molecule, and lubricating oils have up to 30 or more carbon atoms per molecule.

STRAIGHT (OR BRANCHED) CHAIN COMPOUNDS

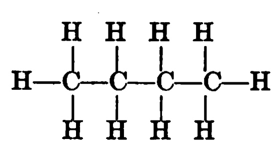
Paraffins.—The paraffin series is composed of molecules with the general formula C_nH_{2n+2}. These compounds have great chemical stability.

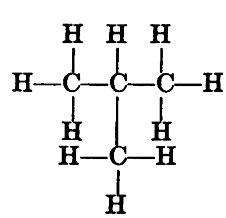
1. Methane 

2. Ethane 

3. Propane 

Starting with Butane, C_4H_{10}, there is more than one possible arrangement, and the branched chain compounds are known as isomers.

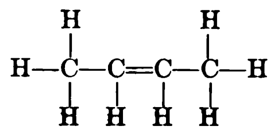
4. Butane 

Iso-butane 

There are 4,347 isomers of $C_{15}H_{32}$

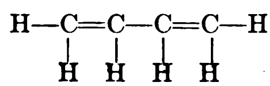
The properties of the isomers differ to a considerable degree for each different arrangement of atoms in a molecule.

Olefins.—The olefins are unsaturated molecules similar to the paraffins except for the presence of a double bond and the corresponding absence of two atoms of hydrogen. The general formula is C_nH_{2n}. The olefins are much less stable than corresponding paraffins. They tend to form gums at high temperatures and are undesirable in lubricating oils.

Butene–2 

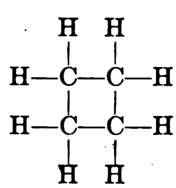
There are 3 butene isomers, and 35,564 isomers of $C_{15}H_{30}$

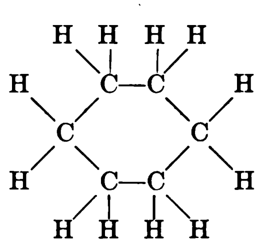
Diolefins.—The diolefins have two double bonds. The general formula is C_nH_{2n-2}. The diolefins are very unstable and form gums even in storage, as in fuel tanks.

Butadiene 

CYCLIC OR RING COMPOUNDS

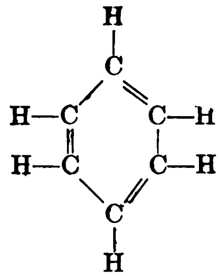
Napthenes.—The napthenes are cyclic compounds with the general formula C_nH_{2n}. They are very different from the olefins, although they have the same number of carbon and hydrogen atoms per molecule. They are relatively stable compounds.

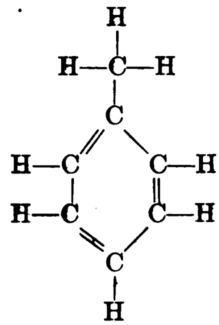
Cyclobutane 

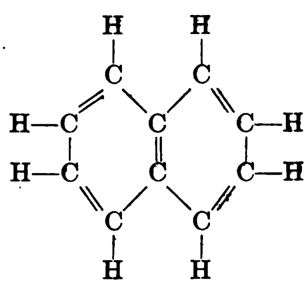
Cyclohexane 

Aromatics.—The aromatic series is composed of unsaturated ring compounds. No general formula can be given. Aromatics make

excellent fuels for spark ignition engines and very poor fuels for compression ignition engines.

Benzene 

Toluene 

Naphthalene 

DENSITY AND SPECIFIC GRAVITY

The density of a substance is its weight per unit of volume. It is customary in American engineering practice to deal with the density in pounds per cubic foot. The density of fresh water at 60° F., 62.4 pounds per cubic foot is a very important constant.

As a matter of convenience, the specific gravity of a substance is often used as an index of its density. The specific gravity of a body is the ratio between the density of the body and that of some substance assumed as standard. For liquids and solids, water at 60° F. is used as the standard substance in America. The specific gravity of water, therefore, is 1.00 at 60° F. It should be written sp. gr. $= 1.00_{60/60}$, indicating that it was at 60° F. when measured, and that it is referred to water at 60° F.

Density and specific gravity are determined by (*a*) finding the mass of a known volume, or by (*b*) finding the volume of a known mass.

There are several laboratory methods of determining specific gravity. Perhaps the most accurate is the use of a pyknometer or specific gravity bottle. This is a small glass bottle with a glass stopper and provided with a vent of very small diameter. The bottle is filled full and the stopper inserted, causing the excess to overflow through the vent. The bottle is then quickly dried and weighed on a precision balance. The net weight of the liquid sample is compared to the net weight of a sample of pure water filling the same bottle, and the specific gravity determined. One of the most rapid precision methods uses the Westphal balance. This is a precision balance fitted with a glass plummet hanging from one arm instead of the usual pan. The weight of the plummet hanging in air is compared to its apparent weight when suspended in a sample of liquid and the buoyant effect of the liquid thus determined. The specific gravity of an unknown liquid is determined by comparing its bouyant effect to that of pure water. The balance is calibrated directly in terms of specific gravity, and need only occasionally be checked against a standard water sample. Neither of these precision methods is suitable for use on shipboard, but a hydrometer is used instead.

A hydrometer is essentially a small sealed flask with a long neck. It is weighted so that it just floats in the liquid to be tested with the liquid surface about half way up the neck. The neck of the hydrometer is calibrated, and a measure of the specific gravity is determined from the point where the upper edge of the liquid meniscus touches the scale.

Hydrometers may be calibrated to any arbitrary scale. Some are calibrated to read specific gravity directly, but those used for petroleum products are more usually calibrated to read specific gravity indirectly. Hydrometers for petroleum products are usually calibrated according to the Bé (Baumé) or the A. P. I. (American Petroleum Institute) scale. There are two scales Bé, one for liquids lighter than water and one for liquids heavier than water. We are concerned only with the former.

Bé and A. P. I. gravities may be converted to specific gravities as follows:

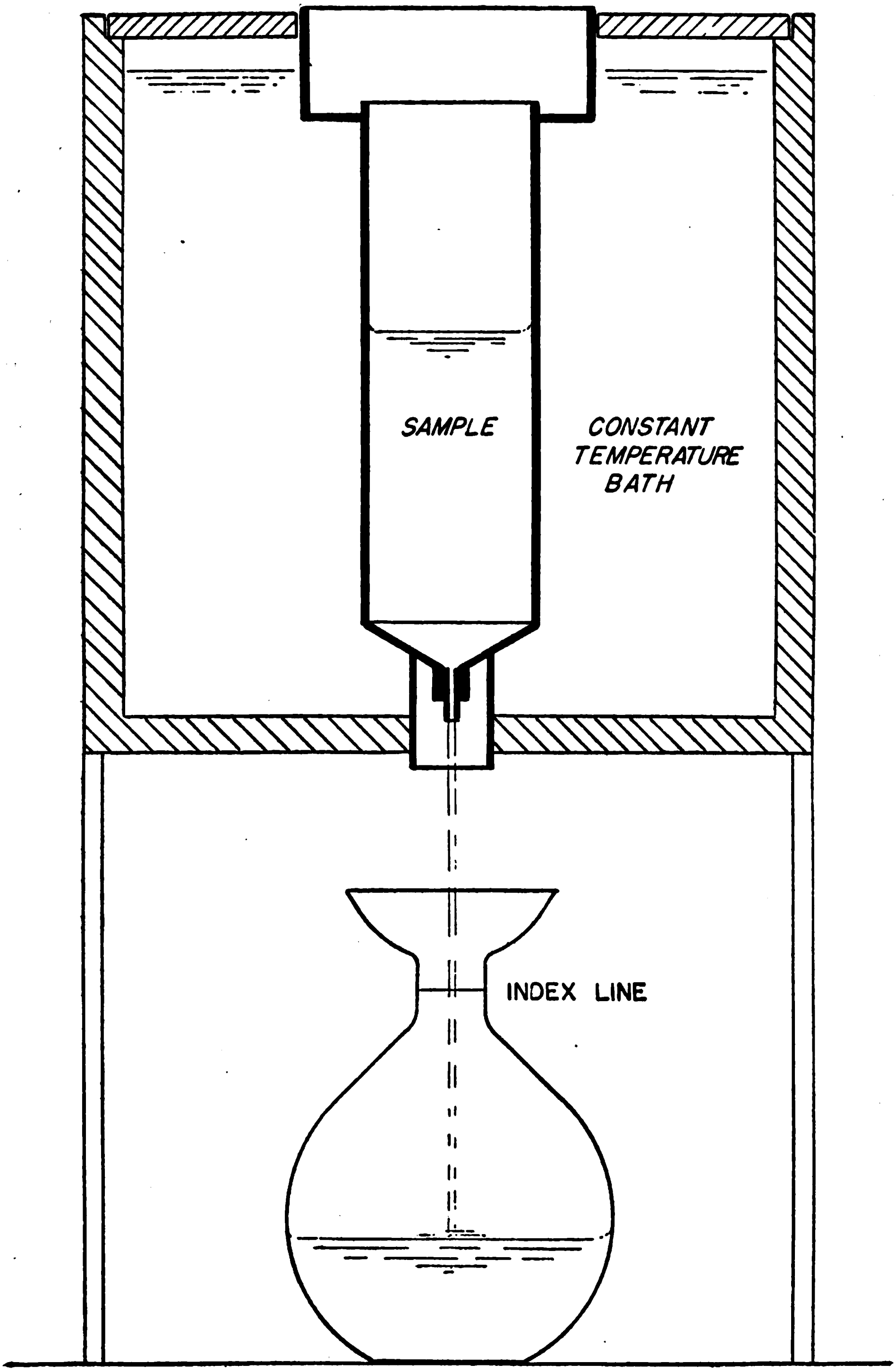
$$\text{Sp. Gr.} = \frac{140}{130 + {}^\circ\text{Bé.}}$$

$$\text{Sp. Gr.} = \frac{141.5}{131.5 + {}^\circ\text{A. P. I.}}$$

A. P. I. gravity is used by the Navy for fuel oil data.

Most petroleum products have about the same coefficient of expansion, so the following rule may be used:

For each 10° F. temperature rise above 60° F., add 0.7° A. P. I.



SAYBOLT VISCOMETER
Figure I.

from the A. P. I. gravity at 60° F., and for each 10° F. decrease below 60° F., subtract 0.7° A. P. I.

VISCOSITY

Viscosity is the measure of the ability of a liquid to resist shear. For reasons which will be apparent later, it is a very important property of lubricating oils. Viscosity may be defined in terms of fluid resistance to flow. Water is a fluid of relatively low viscosity compared to honey, which has a high viscosity.

One of the commonest methods of evaluating viscosity is to allow a standard quantity of fluid to flow under the force of gravity from a standardized container through a standardized orifice at a known temperature. See figure 1. The time in seconds required for the standard quantity of fluid to drain from the container is a measure of the *kinematic viscosity*. No account is taken of the density of the fluid—only of the time required to drain the standard sample. The *absolute viscosity* is obtained by multiplying the kinematic viscosity by the density of the fluid.

Until recently the standard instrument for measuring viscosity was constructed as indicated in the sketch, and was known as the Saybolt Universal Viscosimeter. Results were reported in Saybolt Universal Seconds (S. U. S.)—the time in seconds required to drain the standard cup of liquid from the sample container—at a specified temperature, normally 130° F. or 210° F. Capillary viscosimeters have now replaced the Saybolt instruments, but results are still converted to and reported in S. U. S.

S. A. E. numbers are used to classify lubricating oils for automotive use according to certain viscosity limits.

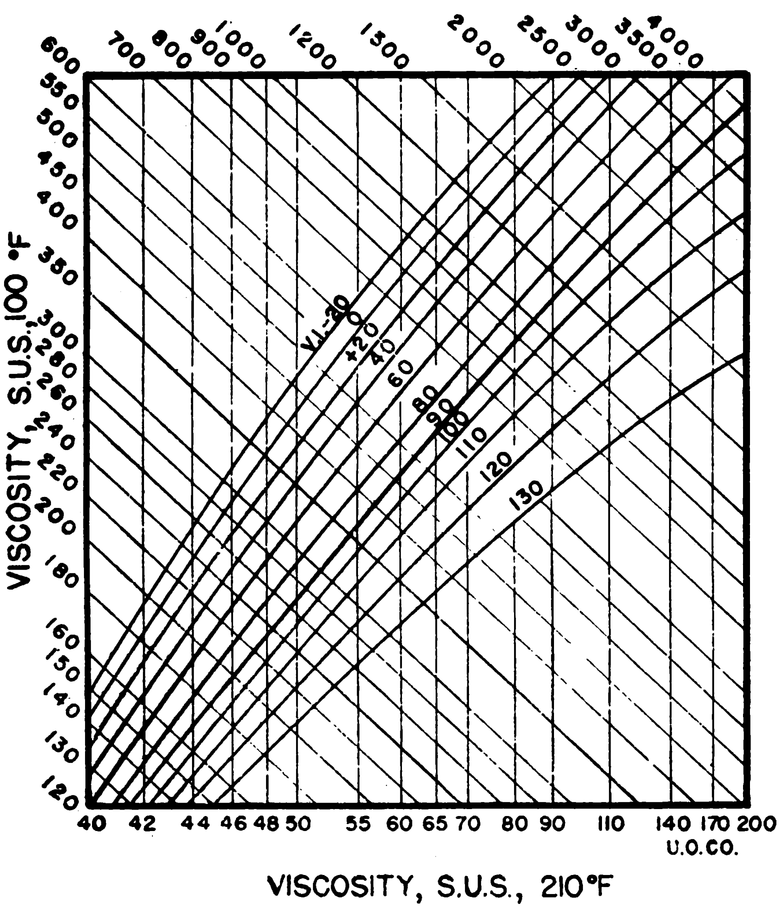
S. A. E. number	Saybolt universal seconds			
	At 130° F.		At 210° F.	
	Minimum	Maximum	Minimum	Maximum
10	90	120		
20	120	185		
30	185	255		
40	255			80
50			80	105
60			105	125
70			125	150

Navy symbol numbers are used to classify lubricating oils meeting the requirements of the Bureau of Engineering. The symbol of identification of lubricating oil consists of four digits, the last three of which give the viscosity in S. U. S., and the first of which designates

the use of the oil and the temperature at which the viscosity is measured. The classes identified by the first digit are:

Forced-feed oils (S. U. S. 130° F.)	symbols 2075 to 2250
Forced-feed oils (S. U. S. 210° F.)	symbols 1080 to 1150 3050 to 3120
Compounded marine-engine oils	symbol 4065
Mineral marine-engine and cylinder oils	symbols 5065, 5150, 5190
Transformer oils	symbol 9045

Viscosity index.—The viscosity of an oil changes rapidly with temperature. The rate of change depends on the type of chemical struc-



VISCOSITY INDEX CHART

Figure 2.

ture of the oil. In order to compare the rate of change of viscosity of various oils, two types of oil were selected as standards of comparison. A paraffin-base oil (Eastern) with a low rate of change was chosen as a high index oil and arbitrarily rated 100 V. I. For the other

extreme a naphthene-base oil with a high rate of change was chosen as a low-index oil and rated 0 V. I. Tables and alignment charts have been prepared so that if the viscosity of an oil sample is known at 100° F. and at 210° F., its viscosity index can be determined. (Fig. 2.) Viscosity indices range from 0 to about 140. Viscosity index is an important property of lubricating oils.

Oiliness.—Oiliness is the property of a lubricant which determines the ability of films a few molecules in thickness to reduce friction. It is not susceptible of direct measurement, but a number of tests are available for determining the relative oiliness of various lubricants.

Other properties.—A number of other properties are obtained by customary laboratory inspection tests.

Sulphur content is an important property of petroleum products, especially fuels. Combustion of sulphur yields sulphur dioxide, which dissolved in water forms sulphurous acid. The presence of acid is serious because it leads to the rapid corrosion of bearing surfaces.

Acidity of petroleum product, sometimes due to the presence of traces of sulphuric acid left from acid treatment during the refining process, is very undesirable, and a low acidity is essential.

The *flash point* of an oil is the lowest temperature at which a flash appears on the oil surface when a test flame is applied under specified test conditions. It is a rough indication of the tendency of the product to vaporize as it is heated.

The *fire point* is the temperature at which the oil ignites and burns for at least 5 seconds under specified test conditions. Its practical significance is questionable.

The *pour point* is determined by noting the highest temperature at which the oil will not flow under specified test conditions.

The Conradson carbon residue is determined by evaporating, under specified test conditions, a known weight of oil and weighing the residue. It was originally intended to indicate the relative tendency of lubricants to form carbon deposits in an engine. It is reported regularly but has not proven entirely reliable for this purpose.

Fuel oil filters.—Fuel oil filters are designed to remove gums and abrasive particles from fuel oil. These filters include cartridge type elements through which the entire flow of fuel oil must pass. Fuel filters are not equipped with spring-loaded relief valves but are generally supplied in a duplex arrangement allowing elements to be replaced without interference in the operation of the engine. Filter cartridges should be replaced with new ones when the pressure drop across the filter reaches the maximum value set by the filter manufacturer.

Lubricating oil filters.—(1) Lubricating oil filters are designed to remove sludge, carbon particles, metallic particles and other forms of foreign material from the oil. They are not designed to remove fuel

oil or water from the lubricating oil. Filters may be arranged to pass the entire flow of oil or only a certain percentage of the oil depending upon the particular installation. In all cases, however, spring loaded relief valves are supplied so that there is no possibility of restricting the flow of oil to an extent that would cause damage to the engine.

(2) Present Bureau approved filters all contain replaceable cartridge type filtering elements. These elements should be renewed when the precipitation number of the oil has reached a value of 0.05.

Strainers.—Strainers consisting of either closely spaced discs or spirally wound wire are used on both fuel and lubricating oil. Strainers for use on fuel oil should have slot spacings of from 0.001″ to 0.0025″ and strainers for use on lubricating oil should have slot spacings of 0.003″. Strainers should be removed and cleaned at the differential pressure or time interval specified in the manufacturer's instructions.

LUBRICATING OILS

(1) Discard the lubricating oil when the fuel dilution becomes great enough to make maintenance of the required pressure impossible or when it becomes excessively viscous from long use or improper cleaning. Keep the filters clean. Centrifuge the oil regularly.

(2) At any time that it is necessary to remove pistons or cylinder liners because of scoring, the crankcase and the lubricating oil system of the engine should be drained and flushed so as to eliminate any particles of metal which may be circulated through the engine in the lubricating oil. This action is taken as a precautionary measure to avoid the possibility of scoring other cylinders of the same engine, or the possibility of the particles finding their way to the main or connecting rod bearings, etc., of the engine.

(3) Special heavy-duty Diesel engine lubricating oils are approved for certain high-speed engines where ring sticking or engine deposits are a serious problem with straight mineral oils. Three grades of heavy-duty oil, complying with Bureau of Ships Specification 14-0-13 (INT), are available.

Navy Symbol 9170 (S. A. E. 20)
9250 (S. A. E. 30)
9370 (S. A. E. 40)

The 9000 series oils on the approved list have been thoroughly tested in all types of Diesel engines available at the Engineering Experiment Station at Annapolis, and service tested by the Forces Afloat. Diesel engine heavy duty lubricating oils are commonly referred to as "additive," "compounded," or "detergent" oils. They consist of base mineral oil to which compounding material has been added. The following instructions apply to the use of these lubricants.

(*a*) *Additive oil.*—The additive agent has the following beneficial effect on the performance of the base lubricant:

(1) It acts as an oxidation inhibitor.

(2) It improves the natural detergent property of the oil; that is, the ability of the oil to remove or prevent accumulation of carbon deposits.

(3) It improves the affinity of the oil for metal surfaces.

(*b*) *Compounded oil.*—The use of a compounded oil in a Diesel engine results in a reduction in ring sticking and gum or varnish formation on the piston and other parts of the engine. In a double-acting engine it reduces the formation of lacquer on the piston rod and hard carbon deposits in the stuffing box.

(*c*) *Detergent oil.*—In dirty engines a detergent oil will gradually remove gummy and carbonaceous deposits. This material, being carried in suspension in the oil, will tend to clog the oil filters in a relatively short time. Normally, a dirty engine will be purged with one or two fillings of the sump depending upon the condition of the engine and quantity of oil used. During the cleaning-up process, the operator should be cautioned to drain the sump and clean the filter, should the oil gage indicate inadequate oil flow.

(4) The efficacy of a compounded oil depends upon the amount of compounding material it contains. This material is consumed in preventing the formation of sludge deposits and it is, therefore, essential that the oil be replenished at reasonable intervals. Optimum drain periods can best be determined by the Forces Afloat on the basis of frequent oil analysis and condition of the engine. The Bureau recommends that drain periods be governed by the following operating limits in order to obtain the most efficient performance:

(*a*) Neutralization number	0.5 maximum.
(*b*) Precipitation number	0.1 maximum.
(*c*) Fuel dilution	5.0 percent maximum.

These limits apply to straight mineral oil as well as compounded oil. These conditions have been chosen so that if emergency operations require exceeding them slightly, no operating difficulties are to be expected. However, it is considered good engineering practice to govern drain periods accordingly.

(5) All navy yard laboratories, as well as the Engineering Experiment Station at Annapolis, are authorized to test used lubricating oils and report analysis promptly to the Forces Afloat. In cases where no analytical data are available, oil drain periods should be governed as follows:

(*a*) For large, slow, and medium speed Diesel engines equipped with absorbent type filters, 500 hours.

(*b*) For large, slow, and medium speed Diesel engines without filters, 300 hours.

(c) For small, high-speed Diesel engines, 100 hours.

(6) All Navy approved oils are miscible. However, to obtain the maximum benefit from additive oils they should not be mixed with straight mineral oils except in emergencies.

(7) Earth type or chemically active filters should not be used with additive oils as the additive will be removed. Filters of the cotton waste, yarn or cellulose type shall be used. The performance of the filter should not be judged by the color of the oil, as detergent oils are dark in color after short periods of use due to finely suspended carbon particles. These particles are not abrasive and are harmless in the oil.

(8) Additive type oil should never be centrifuged with water added. All practicable precautions should be taken to avoid contamination of additive oils with either fresh or salt water, as this contamination will cause emulsification and partial removal of the additive from the oil.

(9) Liquid-filled sight-feed oilers should be filled with distilled water as the minerals present in tap water will form an emulsion with additive oil and rapidly cloud the feed gage, necessitating frequent cleaning. Slight clouding may be expected with distilled water, but this disadvantage is small compared with the benefits obtained from the use of additive oil.

(10) Additive oils on the approved list are not corrosive. Should ground surfaces be found etched or bearings corroded, it is probable that contamination of the lubricant by water or partially burned fuel is responsible. It is important that fuel systems be kept in good repair and adjustment at all times. The presence of water or partially burned fuel in lubricating oil is to be avoided in any case whether straight mineral oil or additive oil is used.

FUELS

INTERNAL COMBUSTION ENGINE FUELS

Fuels for marine internal combustion engines are liquid hydrocarbons. They are usually derived from petroleum. Hydrocarbon fuels may be classed as gasoline (spark-ignition engine fuel) and Diesel fuel (compression-ignition engine fuel). Very different properties are required of the two classes of fuels. The differences are obtained by using hydrocarbons of different chemical composition.

Gasoline is mixed with the combustion air before compression. It must have the property of resisting autoignition due to the heat developed during compression. Since the attainable thermal efficiency is dependent on the pressure reached during compression, and since the compression temperature is dependent on the degree of compression, it follows that the ability of a gasoline to resist autoignition has an important effect on the attainable efficiency of a spark ignition engine.

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The air for a compression-ignition engine is compressed before the fuel is introduced, and hence the fuel has no effect on the degree of compression obtainable. When the fuel is injected it is very important that ignition should take place immediately, and resistance to auto-ignition is very undesirable.

HEATING VALUE

The heating value of a fuel is the amount of heat in B. t. u. which can be obtained from the complete combustion (oxidation) of 1 pound of the fuel. The higher heating value (H. H. V.) is the amount of heat obtainable assuming that the combustion products are cooled to room temperature and the water vapor from the combustion is condensed. The lower heating value (L. H. V.) is the amount of heat obtainable assuming that the combustion products are cooled to some specified temperature without condensing the water vapor.

The specific heats of the gaseous products of combustion are relatively small, so the H. H. V. of a fuel is approximately 960 B. t. u. greater than the L. H. V. per pound of water vapor formed from combustion of a pound of the fuel.

Exact data on heating values are obtained in the laboratory by burning a known weight of the fuel in a calorimeter and measuring the amount of heat developed. A number of empirical equations are available which give the approximate heating value of a fuel. That of Sherman and Kropff

$$\text{H. H. V.} = 18{,}650 + 40\ (\text{A. P. I.}^{\circ}_{60} - 10)\ \text{B. t. u./lb.}$$

is useful when the gravity of the fuel is known. This data can readily be determined if the fuel is available. The Dulong equation

$$\text{H. H. V.} = 14{,}600\ \text{wt\%}\ C + 52{,}000\ \text{wt\%}\ H$$

is useful when the chemical composition of the fuel is known—a relatively unusual case.

PROPERTIES AND COMBUSTION OF GASOLINE

There are two methods of mixing gasoline with the combustion air prior to compression. The most common method is to spray a mist of gasoline into the air as it enters the intake manifold in a device known as a carburetor. This requires a volatile fuel having fractions which evaporate readily so that at least a part of the fuel will evaporate, even when the engine is cold, to produce an explosive mixture that will subsequently ignite readily from the ignition spark. The other method of introducing the gasoline is to spray it directly into the engine cylinder at the start of the compression stroke.

Carburetion is very much more common than injection. In the usual case, the carburetor is much simpler and less expensive than an injection system, and it requires very little maintenance. The big disadvantage of carburetion is that the accurate metering of the fuel delivered to each individual cylinder is not practical. A second important disadvantage is that carburetion requires the use of a very

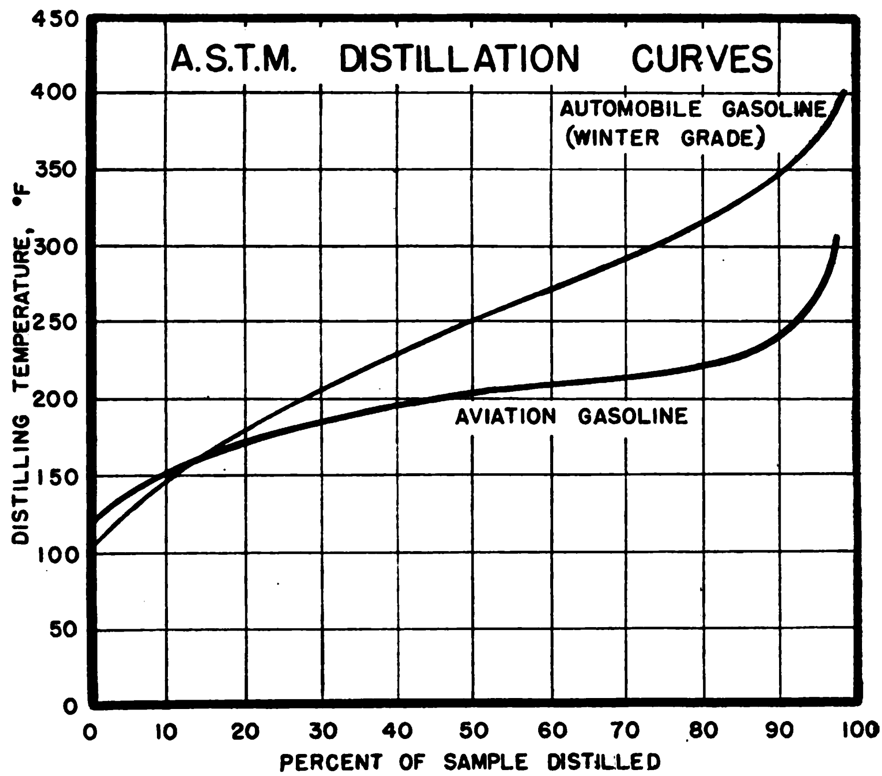


Figure I.

much more volatile fuel than is required by injection, due to the fact that in the latter case the fuel is atomized into droplets a very small fraction of the size of those obtained in a carburetor. The volatility of the fuel is significant because the fire hazard presented by a volatile fuel is many times that presented by a nonvolatile fuel.

Injection is rapidly coming into use on aircraft engines, and with the development of the magnetic nozzle common-rail injection system to be described later injection may compete with carburetion even on automobile engines.

Certain physical properties are desirable in gasoline. It should have a low sulphur content, to minimize acid formation from combustion. It should be nongumming, to reduce the hazard of clogging carburetor orifices. It should have a high heating value, to yield as much power as possible from each pound of fuel carried. It should contain

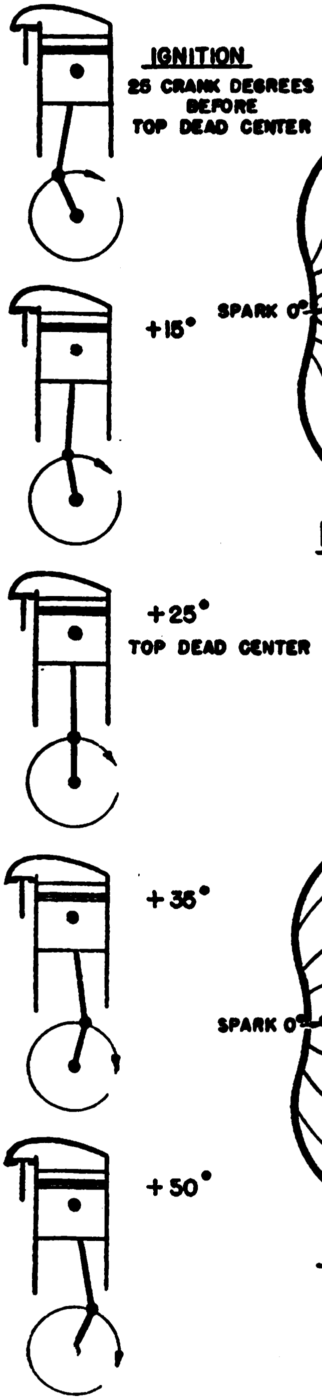
volatile fractions, but the vapor pressure should not be high enough to give vapor lock (clogging fuel lines with bubbles of vapor so that its delivery of liquid gasoline is interrupted). It should not contain a large percentage of high boiling components, particularly if the engine is frequently started and stopped and runs cold, because this may result in serious lubricating oil dilution in the crankcase.

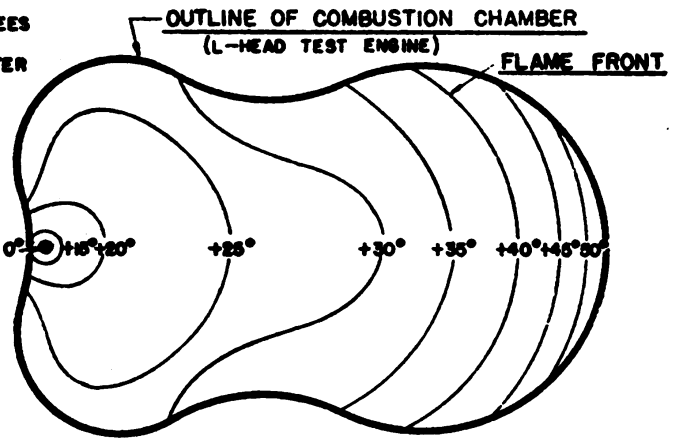
Because of the fact that the volatility of gasoline is important, the A. S. T. M. distillation test has been developed and is regularly conducted as a standard laboratory control test. 100 cc. of the gasoline to be tested is placed in the flask of a specified type of distilling apparatus. The gasoline is heated at a specified rate and the vapor from the flask is led through a condenser cooled with distilled water and shaved ice. The condensate is collected and measured in a covered graduate. A thermometer placed in the neck of the flask with the bulb just below the side arm is used to determine the boiling point of the vapor leaving the flask (which is equal to the temperature of the condensate forming on the thermometer bulb). The temperature at which the first drop of condensate reaches the flask, and the temperature after the distillation of 5 cc., 10 cc., 20 cc., 30 cc. are recorded. From these data A. S. T. M. distillation curves similar to figure 1 are prepared.

There are upper and lower limits of the composition of explosive gasoline vapor and air mixtures. Mixtures containing less gasoline vapor than required by the lower explosive limit will not ignite in the engine. If a gasoline has a high initial boiling point, too little gasoline may vaporize when the motor is cold, and starting would be difficult or impossible. The boiling-point range of the first 20 percent distilled in an A. S. T. M. test is indicative of the starting quality of the fuel in a carbureted engine.

Gasoline with a high boiling point does not vaporize and may enter the engine as a mist even under normal operating temperatures. Some of this mist settles on the cool cylinder walls where it is absorbed by the lubricating oil film. Under the swabbing action of the piston, some of this diluted oil finds its way into the crankcase. Unless the crankcase temperature is high enough to vaporize the diluting gasoline fractions, they will accumulate, and since they have low viscosity and oiliness, they are detrimental to proper lubrication of the engine. The amount of gasoline distilling at temperatures above 350° F. is indicative of the crankcase dilution which may be expected from the fuel.

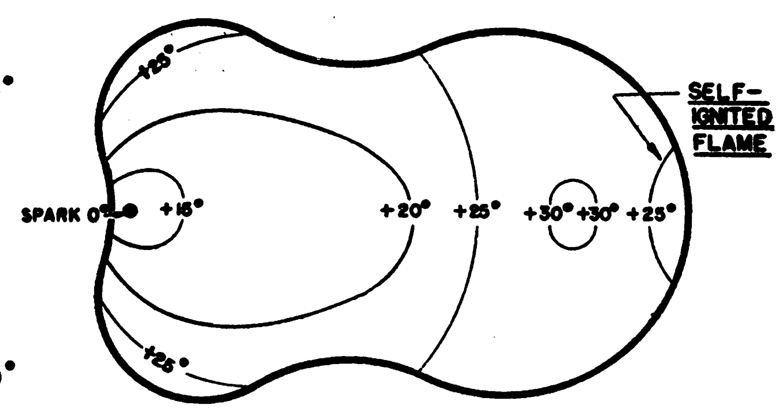
> This does not mean that 350° F. is the maximum acceptable temperature, nor that a large percentage of the fractions boiling above this temperature will work into the oil in the crankcase, but merely that this is a criterion to the relative dilution to be expected from different gasoline under the same operating conditions in a particular engine.





FLAME FRONT PROGRESSION DURING NON-KNOCKING COMBUSTION

Figure 2.



FLAME FRONT PROGRESSION DURING KNOCKING COMBUSTION

Figure 3.

When a combustible mixture is raised to such a temperature that a reaction takes place whereby chemical energy is liberated more rapidly than it can be dissipated by heat transfer, self-ignition will occur. As energy is liberated the temperature rises and the rate of reaction is further increased until the mixture flames and burns.

Under correct operating conditions, the fuel and air mixture in the engine is compressed (to a pressure and temperature limited by the tendency of the fuel toward autoignition), the mixture is ignited by the spark and burns rapidly but smoothly to completion, and the hot gases are expanded, doing useful work on the engine crankshaft. The combustion is not entirely complete until the expansion stroke is well under way. The heat liberated by the combustion is available for use by the engine. A chart of the rate of flame propagation under satisfactory conditions is shown in figure 2. The contours represent the position of the flame front at the stated number of degrees after ignition. Ignition occurred 25° before top dead center.

The combustion takes place during the portion of the piston stroke when the rate of piston movement is low. Consequently there is a rapid rise in the temperature and pressure of the mixture as combustion proceeds. Unless the self-ignition temperature of the fuel is high and unless the ignition delay for the fuel is large, there is a tendency for the flame propagation to proceed very rapidly and for the remaining portion of the fuel to self-ignite and burn instantaneously. This produces a jarring explosion, with an audible knock, and an important part of the energy of combustion is wasted in a hammer-like blow on the engine bearings. Not only is knocking combustion wasteful of a considerable amount of the available power, but the pounding enormously increases wear of the engine bearings and leads to premature failure. The practical limit of compression for a given fuel is therefore the compression which approaches incipient knocking. Flame progression under conditions of knocking combustion is shown in figure 3.

> Engine efficiency increases as the compression ratio is increased up to the point where incipient knocking commences. The rate of increase drops rapidly, and as knocking becomes more severe, the power output drops. If carried to extremes, self-ignition occurs considerably before the end of the compression stroke and the engine is stopped.

The compression at which knocking occurs is dependent on the fuel, the intake air temperature, the engine design (cooling, combustion chamber shape, turbulence, and speed), the degree of spark advance, and the air-to-fuel ratio. There is therefore no precise maximum compression ratio at which all engines may be operated with a given fuel.

Fuels are rated for antiknocking quality in an engine of special design under standard operating conditions. The standard test engine is a variable compression engine designated the C. F. R. (Cooperative Fuel Research) engine. The fuel to be tested is compared to a standard reference fuel, consisting of a mixture of normal heptane (a fuel with low antiknock qualities) and iso-octane, 2,2,4 trimethyl-

pentane (a fuel with high antiknock qualities). The percentage of iso-octane in the mixture is varied until identical knock intensities are obtained with the reference fuel and the fuel under test. The percentage of iso-octane in the reference fuel is reported as the octane number (O. N.) of the fuel under test.

Iso-octane is a synthetic chemical product and is expensive, so in actual practice it is used only to standardize secondary reference fuels which are actually used in routine testing.

A number of chemicals are capable of suppressing autoignition. The outstanding knock suppressent is tetraethyl lead. It acts to

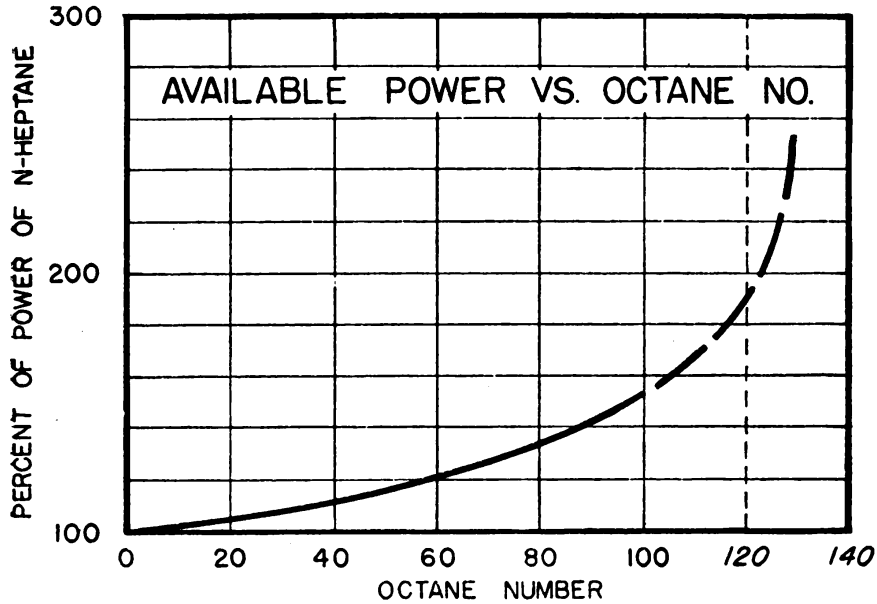


Figure 4.

reduce the rate of flame propagation during the later stages of combustion, without appreciably affecting the burning in the early stages.

At the time that the standards were introduced and adopted, iso-octane was an outstanding antiknock fuel. Since then new methods of refining have been developed, notably with the introduction of alkylation units. The new fuels have a high antiknock rating (about 93 O. N.) and are very susceptible to leading, so that fuels are now available with better antiknock properties than pure iso-octane. There is at present no formally accepted method of rating these super fuels, but the best method seems to be to extrapolate the power-octane scale. This would lead to a maximum octane number of approximately 129, for which (theoretically) the allowable compression ratio would approach infinity and the thermal efficiency would ap-

proach 100 percent. A chart of available power versus octane number, based on unity output by n-heptane, is shown in figure 4. This assumes that the engine is to be operated under optimum conditions for the particular fuel.

PROPERTIES AND COMBUSTION OF DIESEL FUEL

As pointed out in the introduction, Diesel fuel is introduced into the engine after compression of the combustion air, and there is no limit on the thermal efficiency which might theoretically be attained.

The boiling points of Diesel oil fractions are of little importance, since there is no necessity to vaporize a considerable portion of the fuel to obtain a combustible mixture. As in the case of gasoline, it is desirable for Diesel oil to be nongum-forming, and for it to have a low sulphur content. Since one of the important advantages of Diesel fuel over gasoline for marine service is its much lower fire hazard, it is desirable for a Diesel fuel to have a high flash point and a high fire point.

In a properly operating Diesel engine, the combustion air is compressed to a high temperature and pressure, and the injection of fuel is started a few degrees before the crank reaches top dead center. Autoignition of the fuel starts very soon after the fuel is first injected, and continues during the time that fuel is being injected, igniting the fuel as it emerges from the nozzle. Combustion is much slower than in a gasoline engine, and the rate of pressure rise is relatively small.

The first effect on the fuel after injection is partial evaporation, with resultant chilling of the air in the immediate vicinity of each fuel particle. As self-ignition develops, the air and the fuel particles are heated until burning commences. The fuel in the particles is burned as air becomes available, the small particles burning rapidly, and the larger particles taking more time for combustion both because heat must be transferred into the body of the drop to bring it to the ignition temperature, and because the air in the vicinity of the drop is depleted in oxygen and must be replaced. There is always some ignition delay, dependent upon the characteristics of the fuel, the temperature and pressure of the air in the combustion space, the average size of the fuel droplets, and the amount of turbulence present. As combustion progresses, the temperature and pressure of the air and combustion products rises, and hence the ignition delay for newly injected droplets is less than for those injected at first. If the ignition delay is large, a considerable quantity of fuel will collect in the combustion chamber before ignition, and this will burn almost instantaneously once ignition occurs. This produces knock and wasted power.

Attention is called to the fact that ignition lag is the cause of combustion knock in a Diesel engine, but it is the cure for combustion knock in a gasoline engine. An excellent anti-knocking fuel for a gasoline engine is an extremely poor fuel for a Diesel engine, and vice versa. Cooling a gasoline engine reduces the tendency toward self-ignition and reduces combustion knock. Heating a Diesel engine increases the tendency toward self-ignition and reduces combustion knock.

There are three methods in use for rating the antiknock quality of Diesel fuels. The "Diesel index" is determined by a physical chemical test, and is defined by the equation

$$\text{Diesel index} = \frac{\text{aniline point } (^\circ\ \text{F.}) \times \text{A. P. I.}^\circ}{100}$$

The aniline point is the lowest temperature at which equal parts by volume of the test sample and a freshly distilled, water-free sample of aniline are completely miscible. A high Diesel index indicates a fuel of high ignition quality (i. e. one with a low ignition lag). The test is performed by heating the mixture in a jacketed test tube to a clear solution and noting the temperature at which turbidity develops as the mixture is allowed to cool. (The test is indicative of the amount of paraffin-base oil in the test fuel. Straight paraffin oil has a low ignition lag.)

The "critical compression ratio" method makes use of a modified C. F. R. engine that is motored under definite conditions. Fuel is injected for 3 seconds, and if audible combustion occurs the compression is lowered to a compression ratio at which audible combustion can be brought in or eliminated by a definite small change in compression ratio.

The "ignition delay" method makes use of a C. F. R. engine with a special head and electrical apparatus for determining the lag between the start of injection and ignition of the fuel. The compression of the engine is adjusted until a standard delay period is obtained, and the composition of a reference fuel consisting of cetane (a fuel with a very low ignition lag) and alpha-methylnaphthalene (a fuel with a high ignition lag) required to match the ignition lag of the sample under test is determined. The percentage of cetane in the mixture is reported as the *cetane number* of the test sample. This method is being rapidly accepted as the primary antiknock rating test for compression ignition engine fuels.

As in the case of octane number tests, the cetane and alpha-methylnaphthalene are used to standardize secondary reference fuels, and the latter are used for routine testing. Ignition-quality

dopes are sometimes added to Diesel fuel. Among these are alkyl nitrates and nitrites, nitro and nitroso compounds, peroxides, and oxidizing agents. They are not as effective as tetraethyl lead is for gasoline, but the need for them is less acute.

The following tabulation shows approximate relations between the various rating methods:

Cetane number	Octane number	C. C. R.	Diesel index
-------	100	17.4	-------
6	90	14.7	-------
18	70	12.4	0
25	60	11.1	11
50	16	9.0	50
75	(-30)	8.0	90
81	(-42)	7.7	100
100	(-75)	7.0	-------

Chapter VI. THE PRINCIPLES OF LUBRICATION

MECHANISM OF FILM LUBRICATION

Lubrication is provided to reduce friction between moving parts and to prevent metal to metal contact between surfaces moving relative to each other. It also serves the important function of helping to carry away excess heat generated by the friction inevitable between moving parts. Finally, lubricants in some cases serve the additional purpose of sealing piston rings and piston rods gastight.

Bearings may be grouped into two broad classifications—those in which the load is taken between accurately machined surfaces separated by machined precision balls, rollers, or needles (sometimes classed together as "antifriction bearings"), and those in which the load is taken between metal surfaces separated by a film of lubricant. The latter class includes oil-lubricated journals and bearings, oil lubricated thrust bearings, rods and glands, and pistons and cylinders.

In accordance with the thickness of the oil film, the lubrication of oil-lubricated bearings may be split into three general phases:

(1) *Fluid film lubrication* in which the bearing surfaces are separated by a film of lubricant of finite thickness.

(2) *Film lubrication* in which the bearing surfaces are separated by absorbed films a few molecules thick clinging to the journal and bearing surfaces. (The journal is the male part of a cylindrical bearing assembly, and the bearing is the female part.)

(3) *Boundary lubrication* in which the surfaces are lubricated by discontinuous films.

The full lubricating value of a lubricating oil is achieved in film lubrication. The load may be visualized as being carried on rolling molecules of lubricant, and the full oiliness of the lubricant is effective. Increase in film thickness to fluid film thickness offers no opportunity to reduce friction between the bearing surfaces, and it increases the fluid friction resulting from turbulence in the liquid film. The boundary lubrication condition with a discontinuous film is one of imminent failure with some metal to metal contact actually occurring. It therefore follows that from the standpoint of reduction of friction alone that film lubrication with adsorbed films is ideal.

Film lubrication is not practical because of lack of safety factor, and because of the inadequate transfer of heat away from the journal and bearing surfaces. Fluid-film lubrication is the normal and practical condition of oil lubricated bearing operation.

FLUID-FILM LUBRICATION OF JOURNAL BEARINGS

It has been well demonstrated experimentally that bearing loads in fluid-film lubricated bearings are carried by a film of oil under pressure. The pressure is not uniform over the whole bearing area, being atmospheric or slightly below atmospheric pressure (down to the vapor pressure of the lubricant at the operating temperature) on the unloaded sector, and several times the average bearing load near the line of bearing thrust.

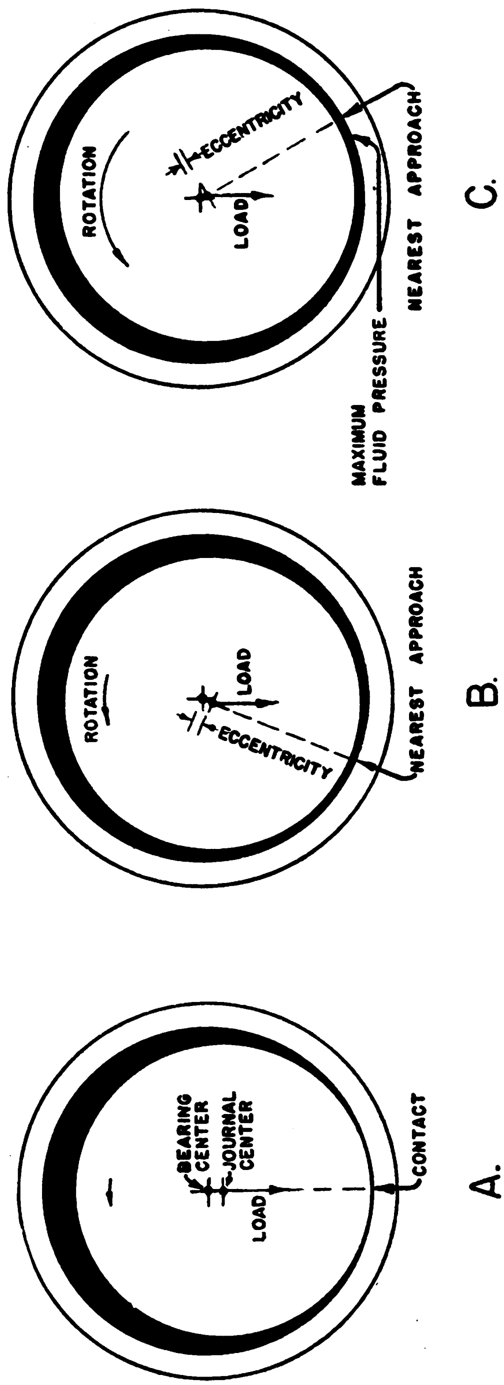
The three sketches in figure 1 show a journal and bearing in three successive positions assumed during starting the journal and bringing it up to speed. Clearance between the journal and bearing is an essential feature of an oil lubricated bearing. The total clearance is generally 0.001 inch for each inch of journal diameter. This clearance is shown, greatly exaggerated, in the sketches.

The load on the journal is acting down in a vertical plane. With the journal at rest, as in A, the journal and the bearing are in contact along a line on the vertical centerline plane of the bearing. Depending on the load on the journal and the properties of the oil, the lubrication along the line of contact is in either the film or the boundary class. As the journal commences to rotate counterclockwise, there is a horizontal component of the friction force that crowds the journal to the left as shown in B. As the journal picks up speed, oil is dragged along by it and a considerable fluid pressure is developed due to the wedge shape of the oil film. This oil pressure becomes great enough to lift the journal clear of the bearing so that it floats on a fluid film. Finally, as the journal comes to full speed, the journal is forced to the right so that the line of nearest approach lies on the side of the centerline plane toward which the journal is rotating.

The journal automatically adjusts its eccentricity and position in the bearing so that exactly the fluid pressure needed to support the journal is developed, providing that the load is within the capacity of the lubricated bearing. Under conditions of high load and low speed, the eccentricity of the journal in the bearing will be a maximum, and the line of nearest approach will lie well up toward the horizontal plane passing through the centerline of the bearing.

Experimental results from a paper by S. A. McKee illustrate these conditions. The data are for a small bearing with a relatively large clearance operating under moderate load and speed. (Actual data: diameter of journal, 0.870 inch; speed, 592 r. p. m.; vis-

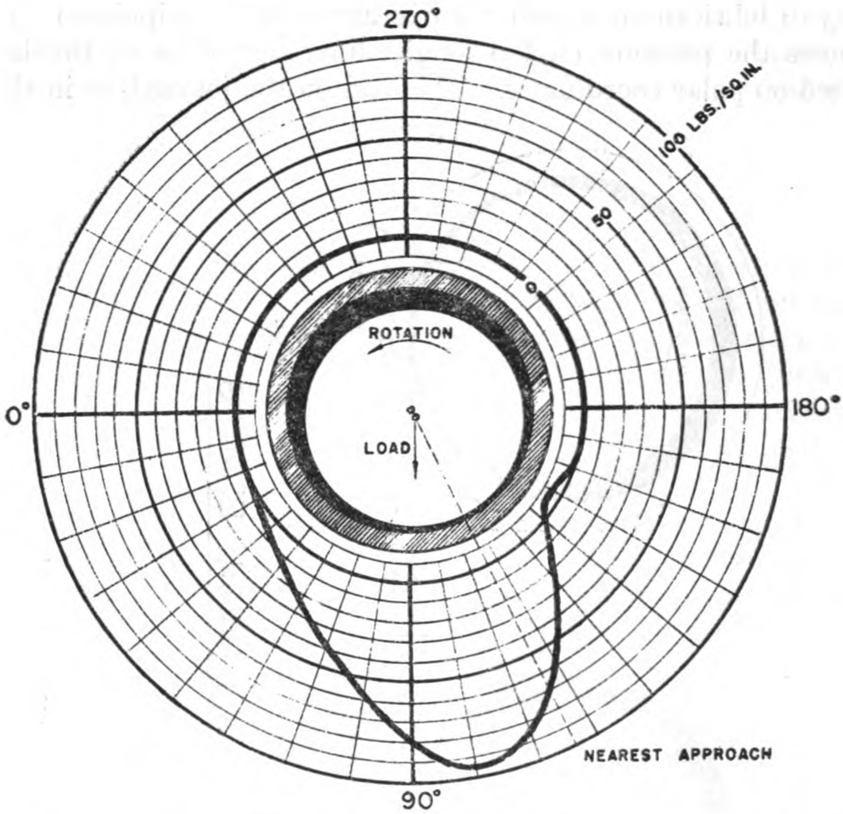
cosity of lubricant at working temperature, 39.4 centipoises.) Figure 2 shows the pressure on the longitudinal centerline of the bearing plotted on polar coordinates. The load on the journal, as in the dia-



OIL LUBRICATION DIAGRAMS FOR JOURNAL BEARING

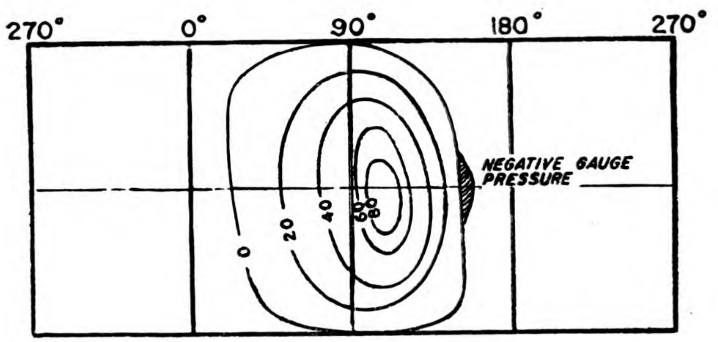
Figure 1.

grams of figure 1, was vertically downward. The direction of rotation also is the same as in figure 1. The region of maximum pressure lies to the right of the bearing centerline as expected, and a region of



POLAR DIAGRAM OF FLUID PRESSURE DEVELOPED AT CENTERLINE PLANE OF A JOURNAL BEARING

Figure 2.



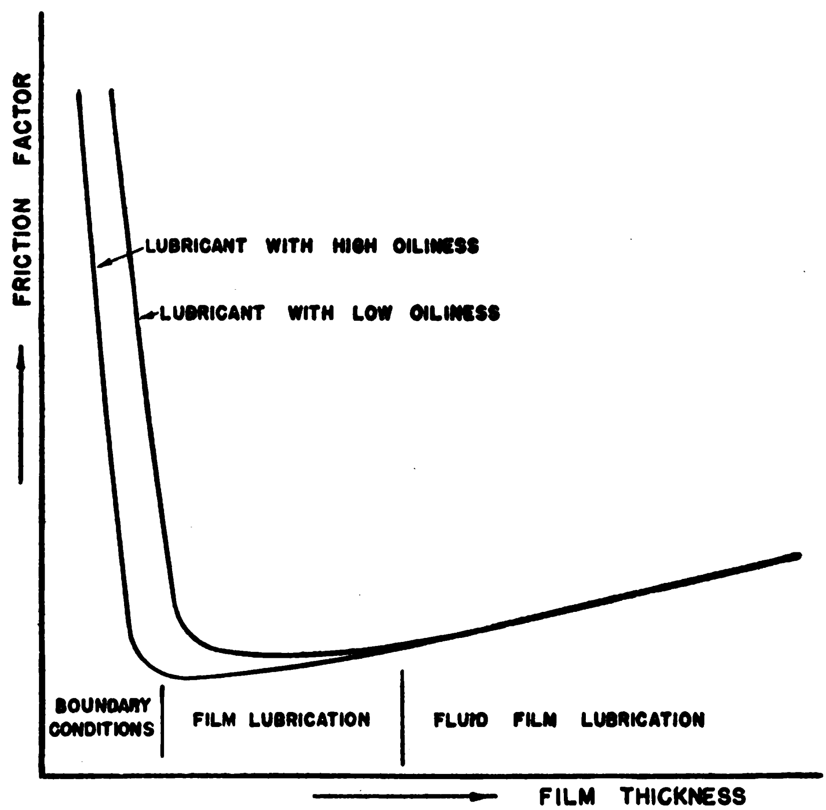
PRESSURE CONTOURS ON BEARING SURFACE

Figure 3.

minimum pressure which is below atmospheric pressure exists a few degrees further in the direction of rotation. Due to the rather large clearance, this region is close to the line of nearest approach. There is no load on the upper half of the bearing shell.

Figure 3 is a development of the surface of the bearing of figure 2 showing lines of equal pressure. The region of negative pressure (shown by hatching) is seen to be restricted to a small area on the centerline of the bearing.

The factors which determine load-carrying capacity are the journal speed (because this determines the rate at which the lubricating oil is dragged into the wedge-shaped space); the clearance between the journal and the bearing relative to the diameter of the journal (because this determines the maximum thickness of the oil film and thus, indirectly, the heat developed from fluid friction, and because the amount of clearance affects the rate of leakage of oil from the pressure longitudinally out of the bearing); the length of the bearing relative to the journal diameter (because this affects the percentage of end leakage); the journal diameter (which together with the speed of rotation determines the surface speed of the journal); and the viscosity of the lubricant at the operating temperature (because this affects the rate at which oil is dragged into the wedge-shaped space at a given journal surface speed).



FRICTION FACTOR VS. OIL FILM THICKNESS

Figure 4.

A coefficient of friction may be expressed as the pounds of force per pound of total load required to obtain a relative motion between two surfaces of indefinite area. Figure 4 is a graph of coefficient of friction vs. oil film thickness for two oils of equal viscosity but of different oiliness. Under normal conditions of bearing operation with a fluid lubricating film, viscosity is the dominant factor. Under these conditions the curves for the two oils are coincident. As the thickness of the lubricating film is reduced, the importance of oiliness increases, and with boundary conditions the oiliness factor is dominant. Oiliness is therefore a factor that enhances the safety of operation in so far as it reduces the generation of heat in the event of failure of the lubricating system. Oiliness is not of appreciable consequence under normal conditions of operation. Oils with high oiliness also reduce wear during starting periods before an adequate oil film has formed for fluid film operation.

For normal bearings, the bearing length is usually not less than one-half nor more than one and one-half times the journal diameter. The maximum length is limited by the rigidity of the journal. The bending of the journal which normally occurs results in a reduction of the film thickness at the ends of the bearing. Unless bearings are short, this reduction in film thickness leads to rupture of the oil film at the ends of the bearing, and finally to local failure of the bearing from excessive heating.

The load carrying capacity of a bearing is proportional to the viscosity of the lubricant at the operating temperature. The heat generated by the operation of the bearing is proportional to the square root of the viscosity of the lubricant at the operating temperature. The decrease in viscosity of an oil with rise in temperature, though dependent on the viscosity index of the oil, is rapid. In many cases the viscosity of the oil in a bearing at the operating temperature is less if an S. A. E. 50 oil is used than it would be with an S. A. E. 30 oil.

Increase in oil temperature promotes oxidation, and for this reason viscous oils usually have a higher carbon residue than low viscosity oils.

A rough bearing or one in poor condition will need a more viscous oil than a smooth, properly fitted bearing, because as the film thickness is reduced there is danger of high spots on the journal rubbing against those on the bearing. If the clearance is unnecessarily large, the oil pressure will be reduced and only an excessively viscous oil will stay between the bearing surfaces. The importance of maintaining designed bearing clearances and of using the grade of oil for which the bearing was designed is therefore obvious.

High speeds permit reduction in viscosity for a given bearing load, since the rotation of the shaft helps to build up the oil-film pressure.

The importance of maintaining the correct lubricating-oil temperature in an oil-circulating system must not be overlooked. All major pressure lubricating systems in naval service are fitted with surface coolers for removing excess heat from the oil, using sea water as the coolant. The connecting oil lines are so arranged that the proportion of the total circulating oil passed through the cooler may be varied at will. By proper manipulation of the system the oil may be maintained at the correct operating temperature.

Fluid-Film Lubrication of Flat Surfaces

The theory of lubrication of flat surfaces parallels that of the lubrication of journal bearings. Parallel surfaces cannot develop fluid pressure due to the relative motion of the parts. A wedge-shaped clearance space is essential to the normal development of fluid-film lubricating conditions. Parallel parts remain separated only by a molecular film of lubricant, and the load-carrying capacity of bearing with parallel surfaces is limited by the load-carrying capacity of the molecular film and by the ability of the parts to conduct the heat away from the bearing surfaces rapidly enough to maintain safe operating temperatures.

High-capacity bearings must provide some means of inclining one of the flat surfaces slightly with respect to the other in order to obtain fluid-film operating conditions. The parts must move in such a direction that the thick edge of the wedge-shaped space between the surfaces is upstream. The Kingsbury thrust bearing is an example of a flat surface bearing in which provision has been made for developing fluid-film lubricating conditions. In this type of bearing the thrust load is distributed between a number of thrust shoes which are mounted on rockers which permit them to assume the optimum angle for carrying the thrust load.

In any flat bearing there is a tendency for the lubricant to flow outward from the center of pressure in all directions toward the edges of the bearing surfaces under the squeezing action of the load applied. If a wedge-shaped clearance space exists between the bearing surfaces, the tendency toward flow is resisted by the viscosity of the lubricant and by the pressure developed in dragging new lubricant into the wedge-shaped space as the result of the relative motion of the parts. An equilibrium condition is developed with the fluid pressure increasing from the edges to the center of pressure, which is normally located on the centerline and toward the training edge of the moving surface. This is shown in figure 5.

Such thrust loads as may be developed in internal combustion engines, as, for instance, by the use of helical timing gears, are small enough to be taken by antifriction bearings or by plain thrust bearings

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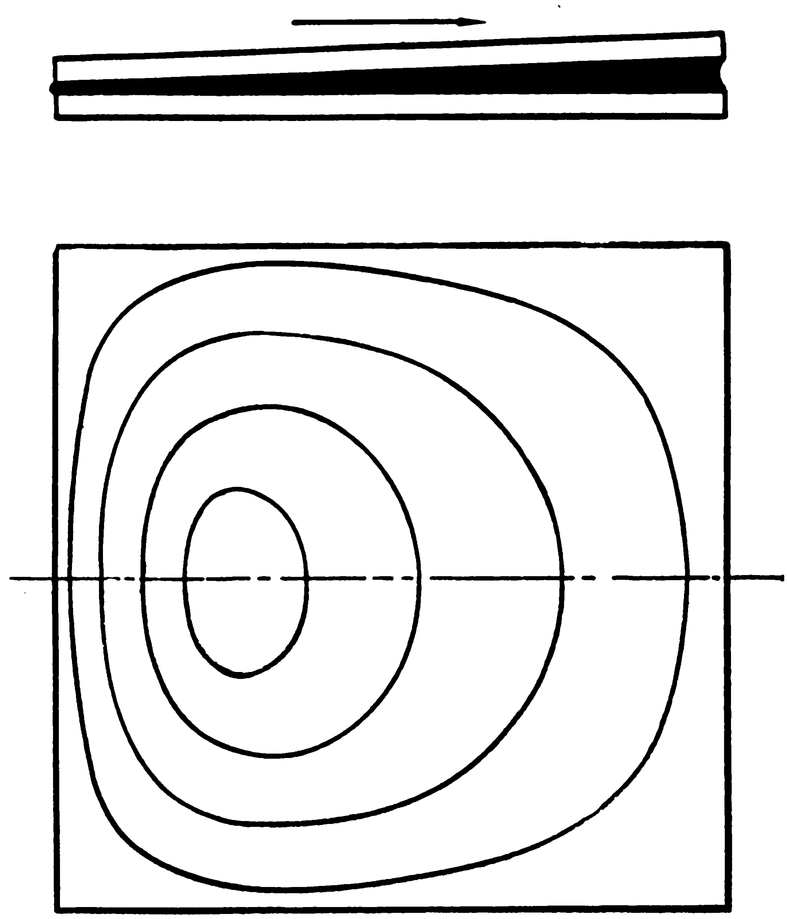
with parallel surfaces. Propellor thrust is usually taken with a Kingsbury thrust bearing.

It has been found by actual measurement that piston rings, initially parallel to the cylinder wall when installed, wear away to present a



PRESSURE CONTOURS ON FLAT BEARING

Figure 5.

double-tapered surface with an angle of about 15 minutes with the cylinder wall, and with the high point approximately 70 percent of the distance from the top to the bottom of the ring. This is illustrated in figure 6. Piston rings are thus capable of forming fluid lubricating films. Lubrication during the breaking-in period, before the rings have developed the tapers, can be greatly assisted by very slightly rounding the edges of the rings.

LUBRICATION WITH GREASES

Greases are compounds of lubricating oil and metallic soaps with an extremely high effective viscosity. They are used where supplying lubricating oil is impractical; where rubbing action must be resisted under conditions of high load and slow relative motion of the bearing surfaces; for the lubrication of antifriction bearings; and where high viscosity and low solubility and emulsibility are required (as in a lubricant for water pump glands).

Most naval machinery is lubricated with fluid oil, and relatively few bearing surfaces are grease lubricated.

OIL GROOVES

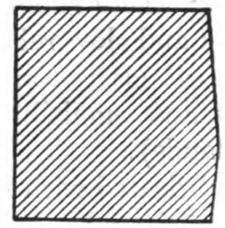
Oil grooves are cut in bearings to assist in the proper distribution of lubricant to the working surfaces.

Because of the fact that the load on an oil-lubricated bearing is carried by fluid pressure built up on the oil film, it naturally follows that anything which permits oil to escape from the high-pressure area will reduce the load-carrying capacity of the bearing. There is probably no better way to relieve the oil pressure in the working part of the bearing than to cut grooves through the working area. Oil grooves must never be cut in the pressure area of a bearing. Complicated and elaborate grooves are not justified, and frequently are so cut that they are actually detrimental. One legitimate exception to this rule is the grooving of bearings at the ends of internal combustion engine crankcases. These are sometimes provided at the outboard ends with circumferential grooves which drain into the crankcase and prevent loss of oil from the engines. Occasionally, in nonreversible internal combustion engines, the end bearings are spiral grooved to pump the oil back into the crankcase. In both of these cases, there is a reduction in bearing capacity which is compensated by using bearings which have adequate effective area.

The legitimate function of oil grooves is to distribute the oil uniformly along the length of the bearing. There are two important rules to follow: (1) cut the oil groove in the unloaded side of the bearing about opposite the line of nearest approach; and (2) eliminate all sharp corners so that there are no edges presented that may scrape the oil from the journal. Grooves should be shallow with well-rounded edges, and should run parallel to the axis of the bearing stopping at least a half-inch from the end of the bearing. Figures 7, 8, and 9, illustrate the principles of correct grooving.

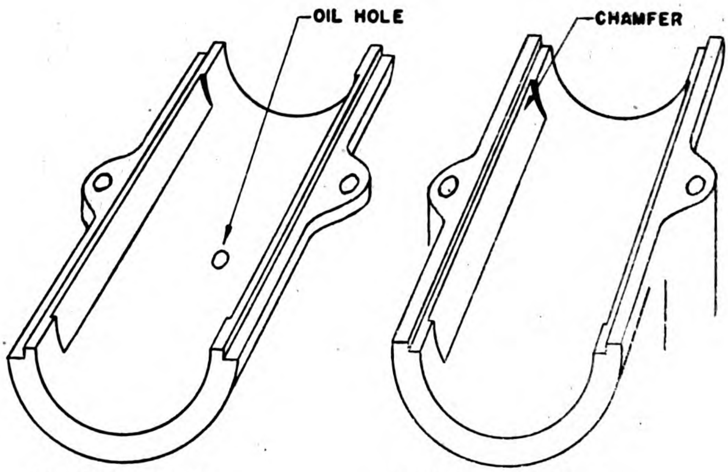
If bearings with split shells are used, they should be installed whenever possible so that the plane of division will be about 90° from the line of nearest approach. When this is done, a shallow chamber with well-rounded edges may be made along each of the four edges of the

bearing halves to within about one-half inch of the ends of the bearing. This will provide a reservoir of oil just ahead of the pressure area. Normally, no other grooving will be necessary, although a longitudinal



SECTION THROUGH PISTON RING SHOWING DOUBLE TAPER EXAGGERATED 20 X.

Figure 6.

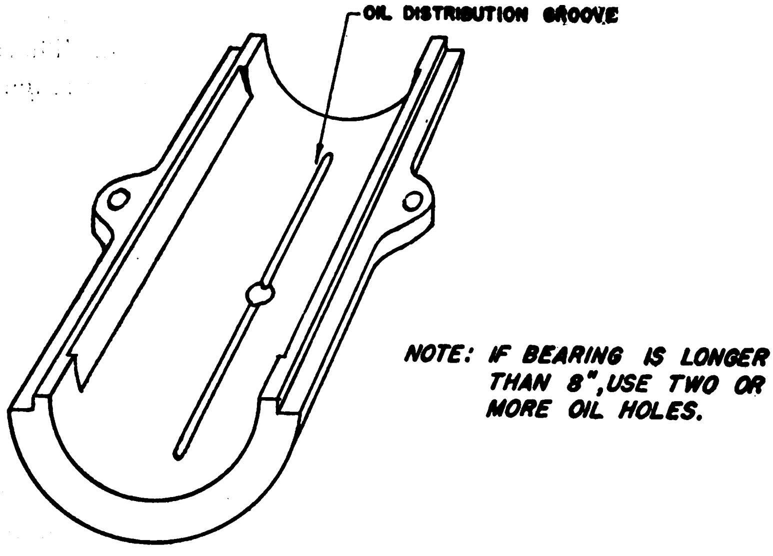


GROOVING OF SMALL BEARING FOR LIGHT LOAD

Figure 7.

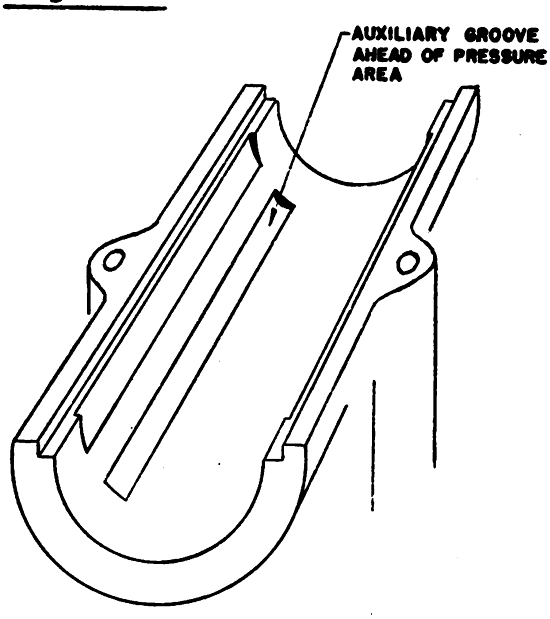
groove at the oil entrance may be cut without appreciably reducing the load-carrying capacity of the bearing.

In the usual case, the center of pressure is a fixed point with respect to the bearing. An exception to this rule is met in the case of crank-pin bearings where the center of pressure is approximately fixed with respect to the crankpin. It is appropriate in this case to groove the



GROOVE ADDED TO CAP OF LONG BEARING WITH MODERATE LOAD

Figure 8.

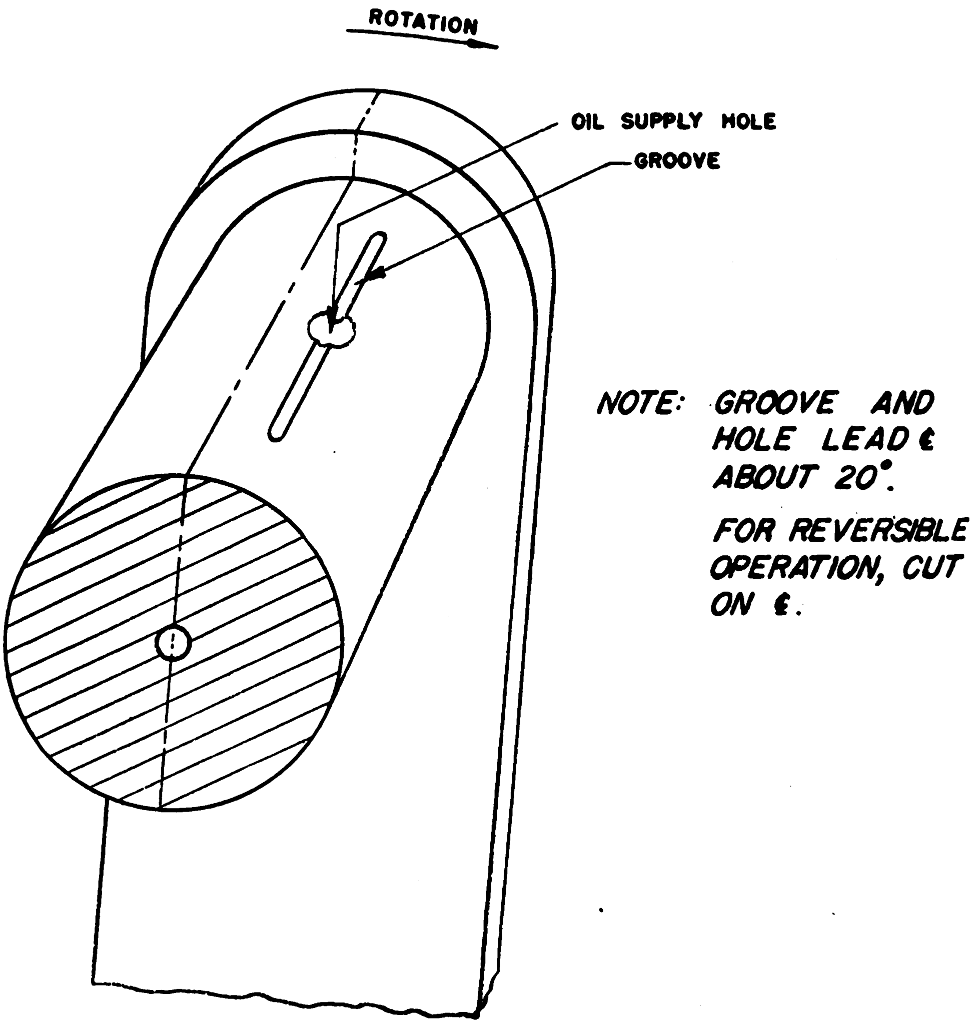


GROOVE ADDED FOR LOW SPEED AND HEAVY LOAD

Figure 9.

journal surface instead of the bearing surface. The groove is so located that the lubricant is supplied at the point having the lowest mean pressure. The principle is shown in figure 10.

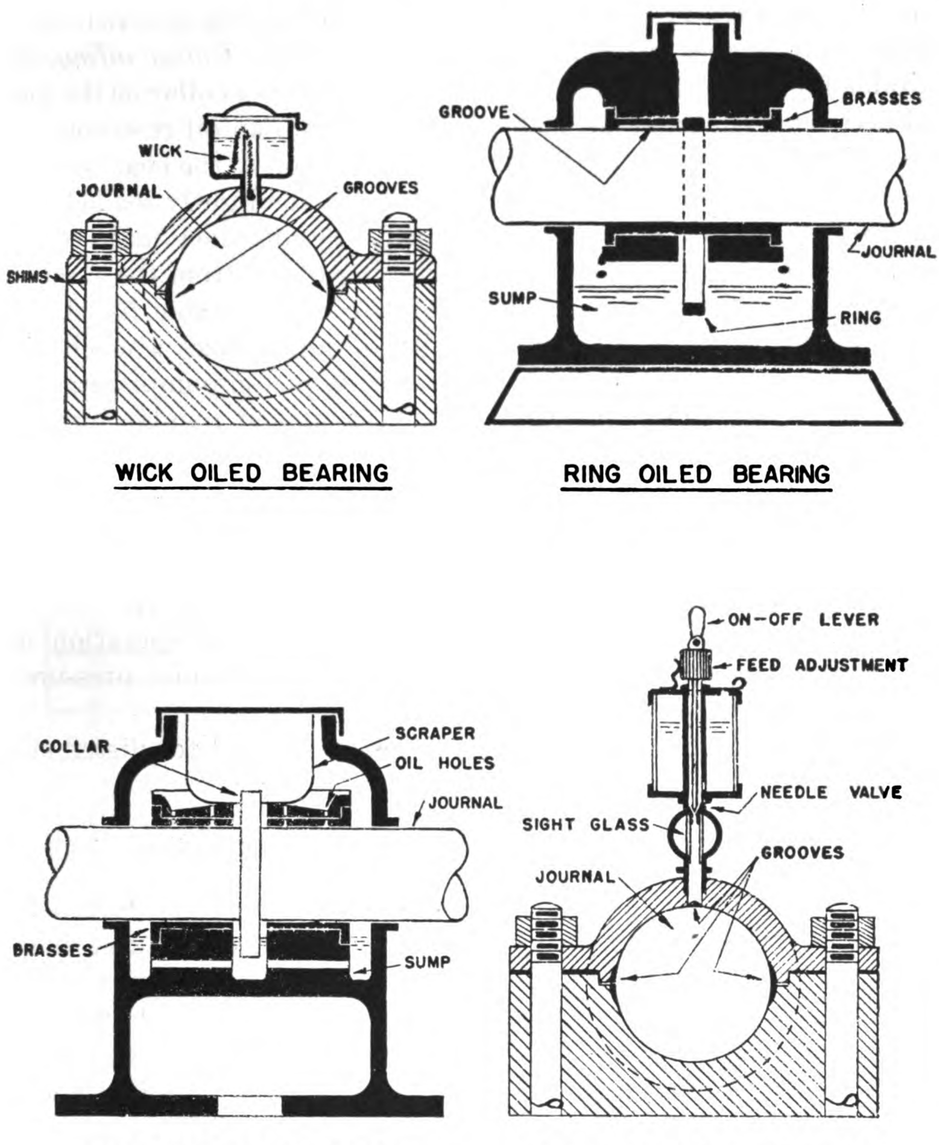
The motion of wrist pin bearings is oscillatory, and the load is not constant in magnitude nor direction of application. There is no opportunity to build up fluid film pressure. In such bearings the function



GROOVE IN CRANKPIN

Figure 10.

of the grooves is to supply lubricant to all parts of the bearing, and more elaborate grooving than is correct for journal bearings may be justified.



LUBRICATING DEVICES

Figure 11.

METHODS OF APPLYING LUBRICANTS

A number of methods are used for supplying lubricant to oil lubricated journal bearings. *Wick oiling*, suitable only for journals operating at low speeds with low unit loading, utilizes an oil-soaked woven wick to supply lubricant from the reservoir to the journal surface. *Ring oiling*, suitable for moderate operating conditions, utilizes a ring riding on the top of the journal through a slot in the bearing cap and

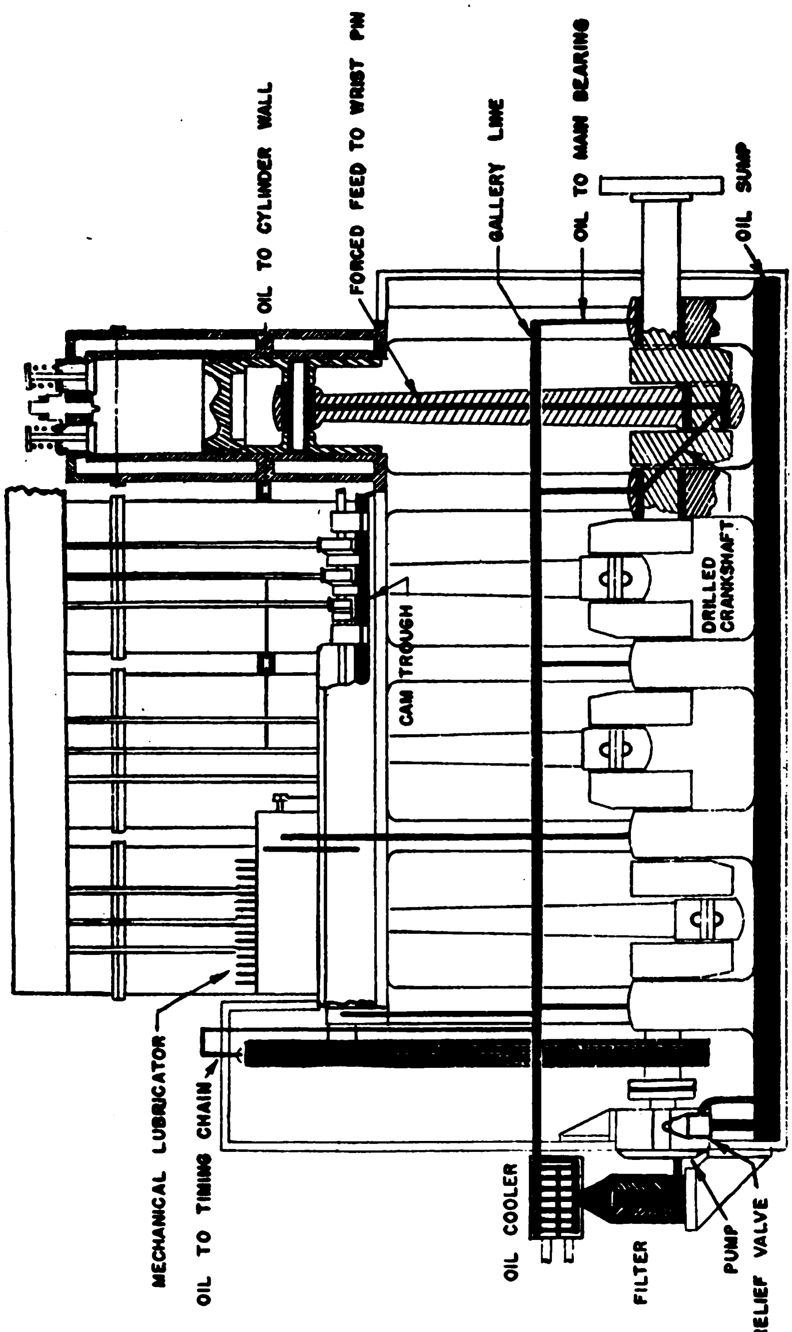
dipping into a reservoir of lubricant under the bearing to which the oil drains as it escapes from the ends of the bearing. *Collar oiling*, also suitable for moderate operating conditions, utilizes a collar on the journal between twin bearings. The collar dips into an oil reservoir, and the oil is removed from the collar for distribution to the bearings with the aid of a scraper. *Gravity oiling*, suitable for lightly loaded bearings, utilizes a gravity-feed oiler which is a container fitted with a needle valve to control the rate of dripping of oil from the reservoir into the bearing supply line. *Mechanical oiling*, suitable for slightly more stringent conditions than gravity oiling, utilizes a mechanical oiler which is a battery of small reciprocating piston pumps supplying a metered quantity of oil to individual bearings at regular intervals. Splash oiling, used for slow reciprocating engines with light bearing loads, utilizes dippers on the ends of the connecting rods of the engine which scoop oil from the crankcase and splash it in all directions. Some of this splashed oil finds its way into the bearings through oil holes located in the unloaded sector of the bearing shells. *Forced feed oiling*, suitable for moderate and severe conditions of operation, utilizes a rotary pump which delivers a stream of oil under pressure to the bearings through suitable piping.

Figure 11 illustrates some of the equipment needed for distributing oil to bearings.

LUBRICATION OF INTERNAL COMBUSTION ENGINES

Nearly all modern internal combustion engines use a full forced-feed lubricating system. Oil is drawn from the pump, which may or may not be the engine crankcase, by a rotary pump. Gear pumps are frequently used. The pump is sized to deliver an excess of oil under normal operating conditions. Constant pressure is maintained on the pump delivery line and the excess oil is bled from this line and returned to the pump by a pressure relief valve. The oil is next passed through a filter and then through an oil cooler. The oil cooler is a surface type cooler, using sea water as the coolant. The filtered and cooled oil is supplied to a gallery line. Branch lines from the gallery line furnish a stream of oil to the main bearing and to timing gear and camshaft bearings. A passage in the connecting rod takes oil from the crankpin bearing and delivers it to the wrist pin bearing. If oil-cooled pistons are used, the coolant is supplied from the wrist pin bearing. Excess oil escaping as a spray from the main bearings, crankpin bearings, and wrist pin bearings, wets all exposed surfaces in the crankcase, assists in cylinder lubrication, and helps to cool the pistons.

A lubricating system is shown diagrammatically in figure 12.



FORCED FEED LUBRICATING SYSTEM

Figure 12.

Oil is supplied to cylinder walls and rings by spray from the ends of pressure lubricated crankpin and piston pin bearings or by splash lubrication. This supply is supplemented for engines having a bore greater than 12 inches by mechanical lubrication through two or more ducts through the cylinder wall. A separate supply pump for each duct is used so that accurate control of the amount and distribution of the lubricant supplied to the cylinder may be maintained.

Lubricant for valve stems is supplied by mechanical lubricators in most instances. Rocker arms are sometimes supplied by mechanical lubricators and sometimes by manual oiling. Cams and cam followers are lubricated from oil picked up by the tips of the cams dipping in oil in a trough installed under the camshaft.

Chapter VII. GASOLINE ENGINES

Except for aircraft service and for PT boat power plants, gasoline engines are of decreasing importance to the Navy. Because of its high volatility and its low flash and fire points, gasoline presents a greater fire hazard than Diesel fuel oil. Diesel engines have been used to replace gasoline engines for naval service wherever their characteristics are suitable, in order to decrease the vulnerability of warships to damage by fire.

In this chapter, the discussion will be limited to features which are peculiar to gasoline engines insofar as this is practical. Features common to gasoline and to Diesel engines will be considered in later assignments.

PERFORMANCE CHARACTERISTICS OF GASOLINE ENGINES

The principal conditions which influence the performance of gasoline engines are: (1) the characteristics of the fuel used; (2) the compression ratio of the engine; (3) the pressure and temperature of the charge in the engine cylinder at the start of compression; (4) the air to fuel ratio of the mixture delivered to the engine; (5) the ignition timing; (6) the engine design, including the combustion chamber design, the location and the number of spark plugs in the cylinders, the valve arrangement, the size of the valves, the manifold arrangement, and the presence or absence of hot spots; and (7) the engine operating conditions including speed, constancy of load and speed, the characteristics of the lubricants, and the operating temperatures. A complete analysis of these factors, most of which are interdependent, is an exacting and complicated task. This discussion will be limited to an outline of a few important factors.

Definitions.—The brake mean effective pressure is the apparent net average pressure acting on the piston during the working stroke; specifically it is defined by the equation.

$$\text{B. m. e. p.}=\frac{\text{Net work}}{\text{Displacement}}=\frac{W}{LAN}\times 33{,}000 \text{ (lb./sq. in.)}$$

W=Net work output—brake horsepower.
L=Piston stroke—feet.
A=Piston area—square inches.
N=Power strokes per minute.

Brake mean effective pressure is used for comparing the performance of engines because it eliminates the major part of the effects of the size and speed of operation of the engine and of the piston strokes per cycle.

The *air to fuel ratio* is the ratio of the pounds of air per pound of fuel in the charge supplied to the engine. A *lean mixture* is one with a large air to fuel ratio, and a *rich mixture* is one with a small air to fuel ratio.

The *economy* of an engine is measured by the pounds of fuel required per brake horsepower (the actual power output of the engine) in a given time interval. A low numerical value of economy is indicative of economical operation of the engine.

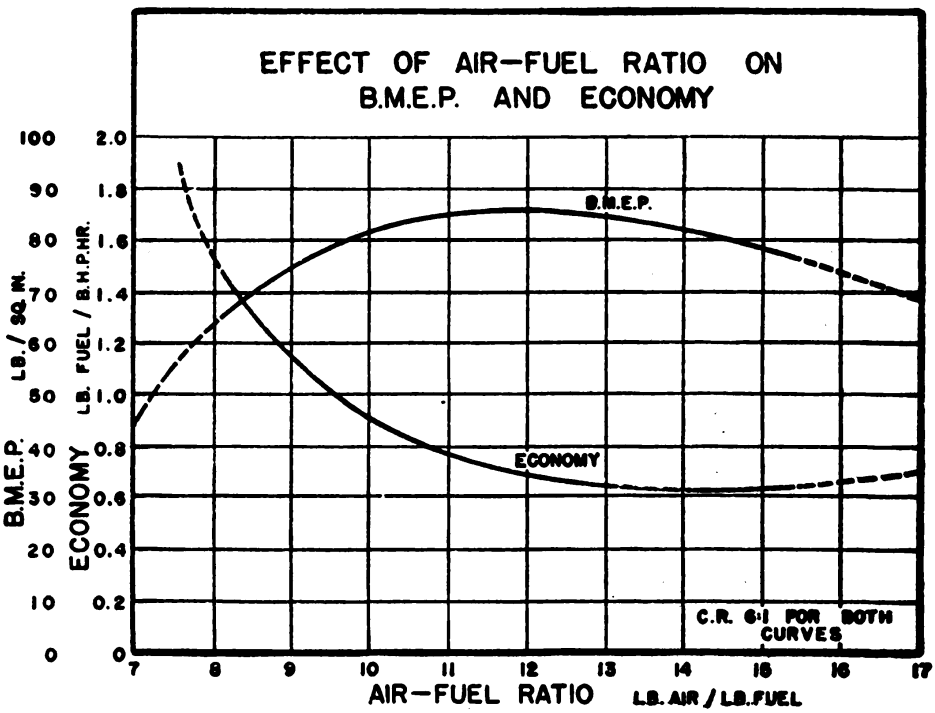


Figure 1.

Effect of air to fuel ratio on engine performance.—The effect on b. m. e. p. and on economy of changing the air to fuel ratio is shown in figure 1 for a gasoline engine operated at constant speed. This engine was operated on unleaded automobile gasoline with a compression ratio of 6.0:1.0. Note that with very rich mixtures the power output of the engine is low and the economy is poor. Maximum power is obtained with a richer mixture than is required for maximum economy. Operation at maximum power instead of at optimum economy is attended by a 10-percent loss of economy for a 5-percent gain in power. Note also that very lean mixtures are uneconomical.

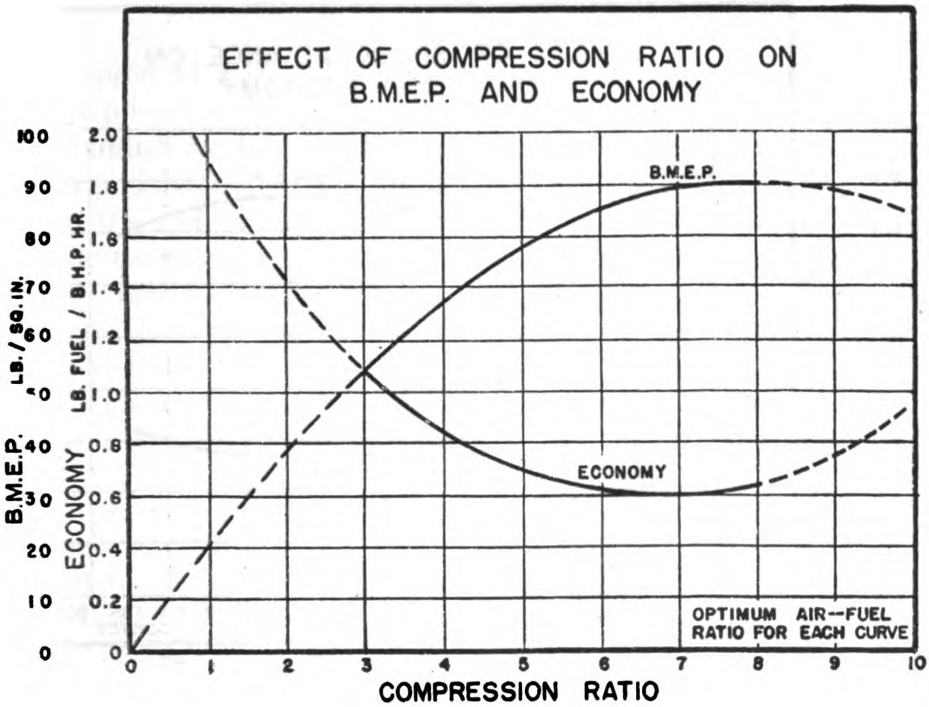


Figure 2.

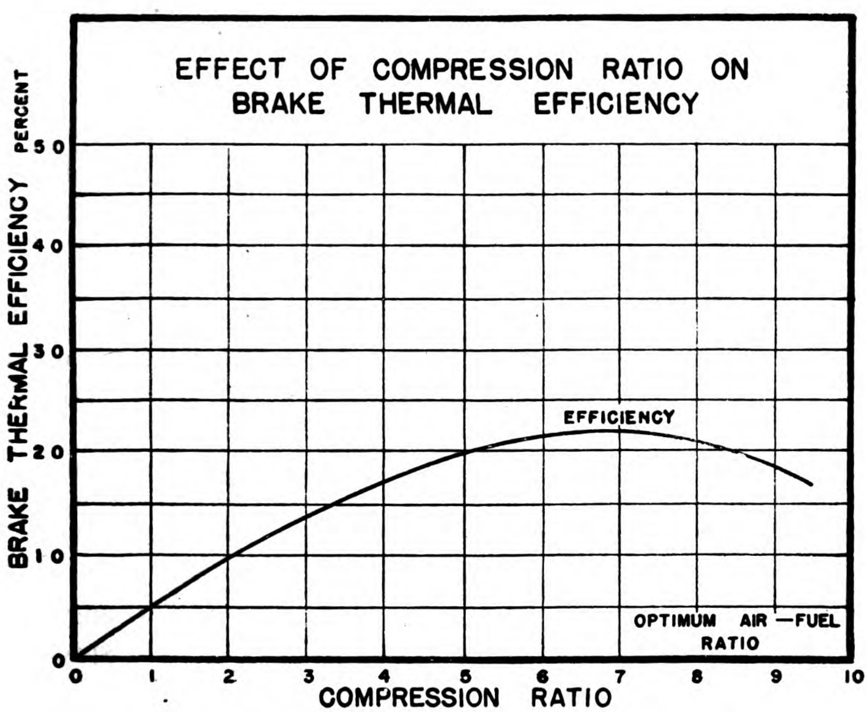


Figure 3.

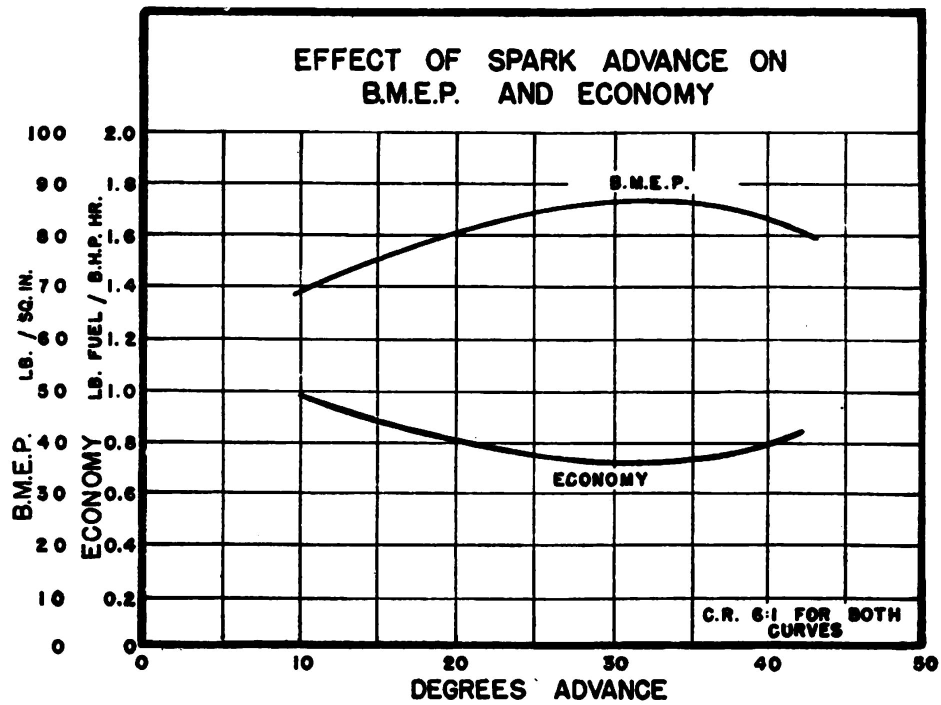


Figure 4.

Effect of compression ratio on engine performance.—The effect on optimum economy and on maximum power output of changing the compression ratio is shown in figure 2. The engine and the fuel were the ones that were used to obtain the data for figure 1. The maximum usable compression ratio is limited by detonation of the fuel. Attempting to exceed this maximum leads to severe knocking and to loss of efficiency as indicated by the dotted portions of the curve.

The data on economy from figure 2 have been reduced to brake thermal efficiency and plotted in figure 3. The Otto cycle curve of figure 8, Thermodynamics 4, is the theoretical equivalent of the curve of figure 3.

Effect of spark setting on engine performance.—In figure 4 the effect on b. m. e. p. and on economy of changing the timing of the spark with respect to the position of the engine crank is shown. The spark advance is the number of degrees the crank must turn to reach top dead center after the ignition spark occurs. The engine was operated on unleaded automobile gasoline at constant speed with a compression ratio of 6.0 : 1.0.

ADJUSTING THE SPEED OF A GASOLINE ENGINE

The speed of a gasoline engine is normally altered by throttling. A valve placed at the entrance to the intake manifold is manipulated to accomplish throttling. Closing the valve (1) reduces the amount of charge delivered to the engine cylinders; (2) decreases the pressure of the charge in the cylinder at the start of compression and hence decreases the maximum compression and the thermal efficiency; and (3) increases the work required to draw the charge into the cylinder. Each of these effects acts to reduce the amount of power available from the engine, and hence tends to reduce the engine speed. The throttle is usually incorporated as a component part of the carburetor assembly.

;ARBURETORS

A carburetor is a device used on many gasoline engines to mix air and fuel for delivery to the engine cylinders through the intake manifold. It provides a means of spraying a mist of fuel into the air, and controls the air to fuel ratio.

The usual operation of gasoline engines requires that the carburetor should be adjusted to meet the following conditions:

(1) Furnish a definite amount of moderately rich mixture at the minimum throttle setting so that idling of the engine will be smooth and positive.

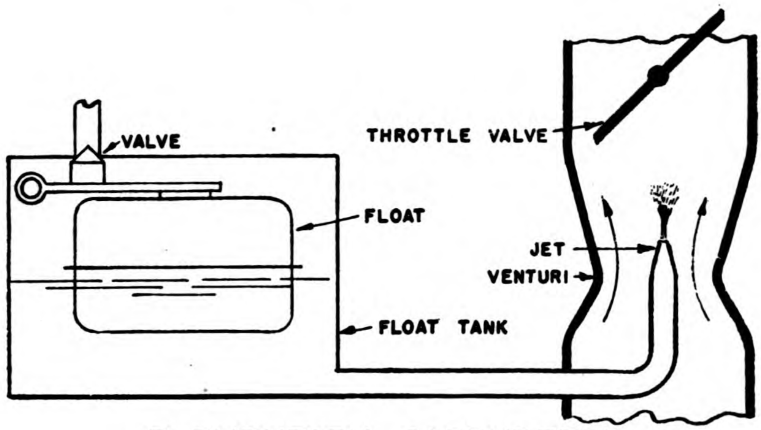
(2) Supply an economical (lean) mixture to the engine at normal operating speeds.

(3) Supply a mixture for maximum power (rich) at full throttle.

(4) Furnish a rich mixture for a short time when sudden acceleration is required.

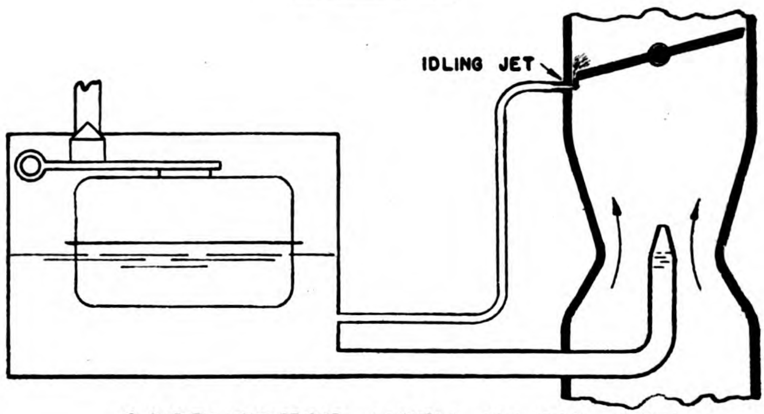
Gasoline engines accelerate slowly with a lean mixture. Sudden opening of the throttle valve allows the pressure in the manifold to rise sharply, and this in turn causes the heavier fractions of the gasoline to condense on the manifold walls and momentarily makes the mixture reaching the engine leaner than that delivered by the carburetor. As the wall film builds up the mixture tends to return to normal. These factors, together, make it desirable to supply a rich mixture for a short time during acceleration.

A fluid flowing at a steady rate, adiabatically, through a channel with a section of restricted cross sectional area must flow at a higher velocity in the restricted section than it does in the unrestricted part of the channel. This increased velocity is attended by a higher kinetic energy which must be obtained from a reduction in potential energy at the section, and this is manifested by a *decrease in pressure*. A



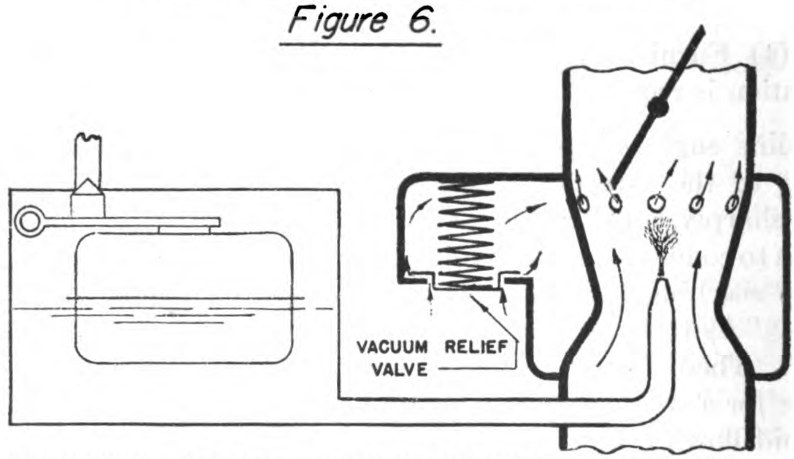
ELEMENTARY CARBURETOR

Figure 5.



CARBURETOR WITH IDLING JET

Figure 6.



VACUUM RELIEF COMPENSATED CARBURETOR

Figure 7.

smoothly rounded restricting and reexpanding section is a Venturi section of a channel. The Venturi effect is used in carburetors to aspirate fuel from the carburetor jet nozzles.

> Except for slight friction and turbulence losses, no energy is wasted in a well-designed Venturi, and the fluid returns to its original velocity and pressure as it returns to the unrestricted channel.

An elementary carburetor is shown diagramatically in figure 5. Its basic features are a *float tank* in which a constant fluid level is maintained by a float operated valve, a *jet*, a *Venturi*, and a butterfly *throttle valve*. The tip of the jet is above the fuel level in the float tank so that when the engine is stopped no fuel will flow from the nozzle. At normal engine operating speeds, the flow of air through the Venturi section produces a pressure drop resulting in a partial vacuum that draws fuel from the jet.

When the engine is operating at low speeds, the vacuum in the Venturi section is not great enough to draw fuel from the main jet. A special jet, termed an *idling jet*, is installed at the edge of the butterfly valve where the velocity is locally high enough to draw fuel from the jet. An idling jet is shown in figure 6. Under normal operating conditions at moderate speeds the velocity at the edge of the butterfly valve is too small to draw fuel from the idling jet, so this jet functions only at idling speeds.

Operation of the engine at higher speeds produces a relatively high vacuum in the Venturi section and would draw a large amount of fuel from a simple jet. At the same time the vacuum rarefies the air in the Venturi. The result is that a simple carburetor would give too rich a mixture at high speeds. Various compensating devices are used to counteract this tendency. Figure 7 shows the principle of a design in which the rich mixture is diluted with air drawn into the carburetor between the engine and the Venturi throat through a *vacuum relief valve*, normally held closed by a light spring. Figure 8 shows a compound or *air-bled jet* provided with a metering orifice for controlling the supply of fuel to the jet and allowing air to enter the line to the jet to relieve the vacuum on the metering jet when the engine is operated at high speed. Still another type of compensation employs a needle valve operated by the throttle to restrict the delivery of fuel from the main jet to the desired amount. Carburetors are frequently supplied with more than one jet, one of which is a simple jet and another which is compensated.

A carburetor with an idling jet and some sort of main jet compensation fulfills requirements of positive idling and economical operation at moderate throttle settings. To provide a richer mixture for

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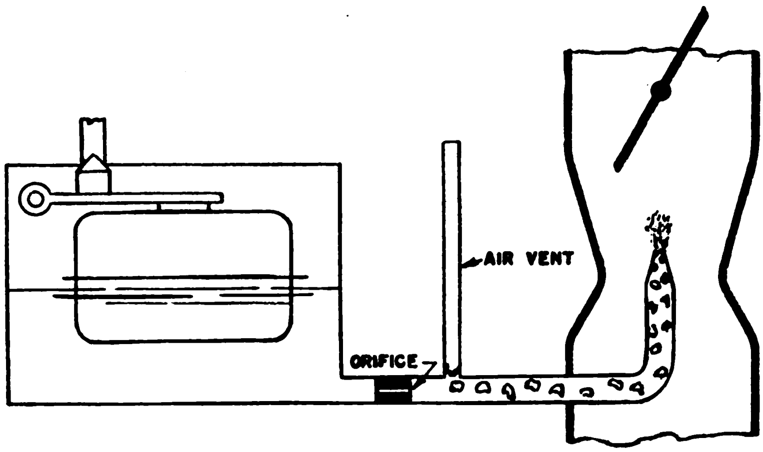
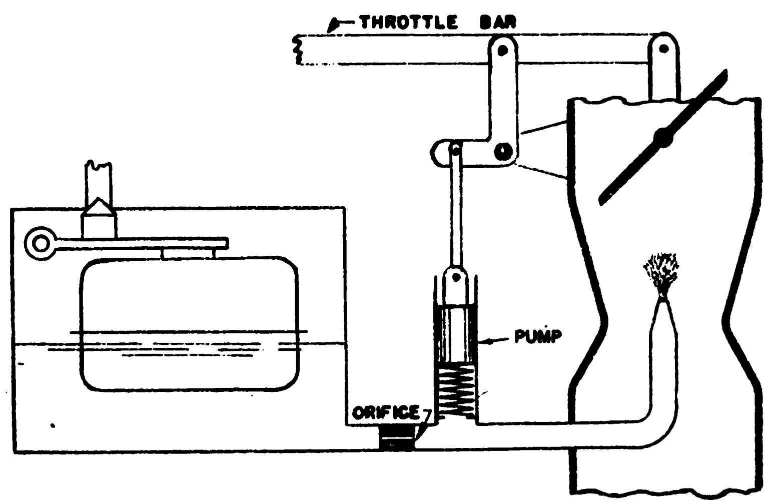
maximum power at full throttle, an additional jet, known as an *economizer jet*, is provided. The supply line to this jet is closed by a valve which is opened when the butterfly valve is turned to full throttle position.



COMPENSATED CARBURETOR WITH AIR BLED JET

Figure 8.



CARBURETOR WITH ACCELERATOR PUMP

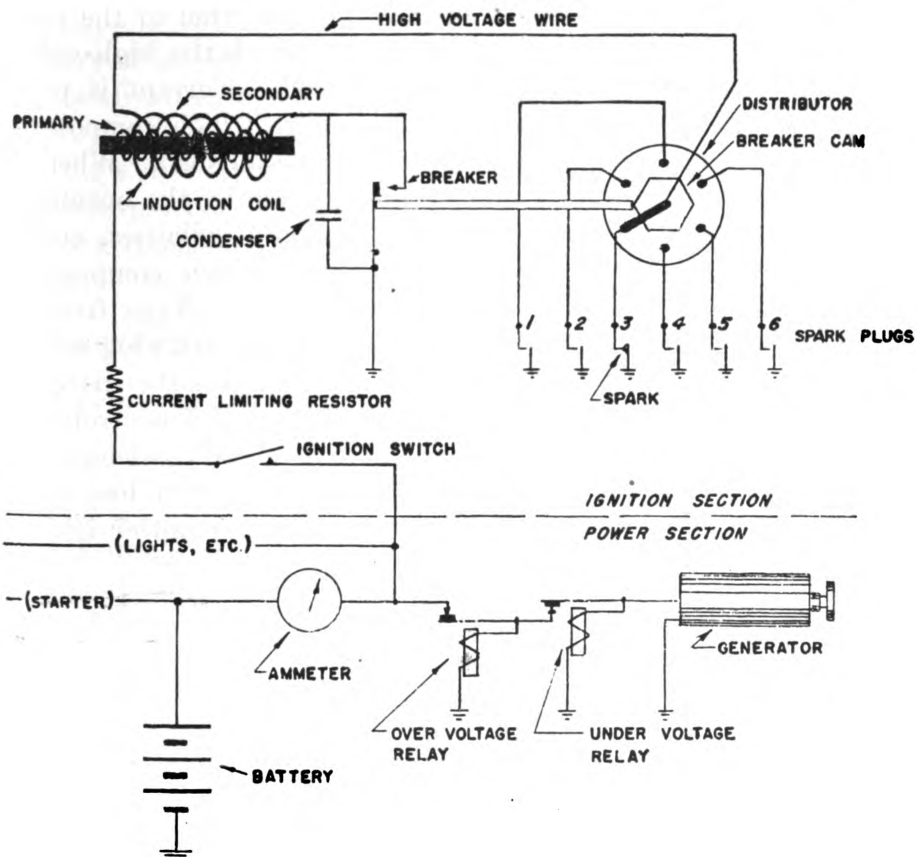
Figure 9.

A surcharge of fuel is provided for sudden acceleration by an accelerator pump. Such a pump is shown in figure 9. A charge of gasoline is contained in a cylinder with a loosely fitted piston. Sudden opening of the throttle compresses the spring which drives the piston downward and delivers the fuel in the accelerator cylinder to the jet.

Some fuel leaks past the piston, but the rate of leakage is not great enough to offset the rapid downward motion of the piston. If, however, the throttle is opened slowly, the piston will be forced down slowly and the fuel in the piston will have time to leak past the piston so that no surcharge will be delivered to the engine.

IGNITION SYSTEMS

The function of an ignition system is to initiate combustion at the proper instant near the end of the compression stroke. Ignition is accomplished in gasoline engines by passing a spark between the elec-



BATTERY IGNITION SYSTEM

Figure 10.

trodes of a *spark plug* at the proper instant in the engine cycle. The construction of a typical spark plug is shown in the chapter Engine Fundamentals.

A battery-type ignition system is shown in figure 10. The circuit has been divided into a power section and an ignition section. The power system consists of a generator driven by the engine, under- and

over-voltage relays, an ammeter, and a battery. The function of the generator is to produce low voltage direct electric current. The under-voltage relay disconnects the generator from the battery when the output voltage of the generator is below battery voltage, so that current cannot be drained from the battery through the generator. The over-voltage relay disconnects the battery from the generator when the generator output voltage is high enough to be injurious to the battery. The ammeter enables the operator to check the functioning of the system.

The ignition section consists of an on-off switch, a current limiting resistor for the protection of the coil, an induction coil, breaker, and condenser, distributor, and spark plugs. The function of the induction coil with its associated equipment is to furnish the high voltage current surge required for each ignition spark. Current is passed through the few turns of relatively large wire which compose the primary of the coil, and a magnetic field is established. When the flow of current through the primary is interrupted by the opening of the breaker points, the magnetic field suddenly collapses, cutting through the large number of turns of wire which compose the secondary coil, and producing a very high output voltage from the secondary circuit. This high voltage is led to the spark plug selected by the distributor, and the energy is discharged across the spark gap to the circuit ground. The operation of the breaker is controlled by a cam on the distributor shaft. The function of the condenser is to protect the breaker points, and, by decreasing the power loss due to arcing at the breaker points, to aid in producing a hotter ignition spark.

Chapter VIII. DIESEL ENGINES

Diesel engines are compression ignition engines with a high-compression ratio. The compression temperature is high enough to raise the fuel above its auto-ignition temperature as soon as it is sprayed into the combustion chamber. Diesel engines are used in the Navy (1) for powerboat propulsion; (2) for driving auxiliaries on all classes of ships; and (3) as the main engines of submarines, some classes of auxiliary ships, small fighting craft, and ships of the train. They have the advantages of (1) high over-all efficiency at both full and part load, (2) being independent, compact, self-contained power units, (3) requiring a minimum of operating personnel, and (4) using inexpensive, nonexplosive fuel (in comparison with gasoline). Because of the high over-all efficiency of Diesel engines, motorships have a large cruising radius relative to their bunker capacity.

1. PERFORMANCE CHARACTERISTICS OF DIESEL ENGINES

The principal conditions which influence the performance of Diesel engines are similar to those affecting the operation of gasoline engines. They are (1) fuel characteristics; (2) compression ratio; (3) the pressure and temperature of the air in the engine cylinder at the start of compression; (3) the quantity of fuel injected per stroke; (4) the injection timing; (5) the injection rate; (6) the completeness of atomization of the injected fuel; (7) the engine design, including the combustion chamber design, the valve arrangement, the size of valves, the manifold arrangement, and the presence or absence of hot spots; and (8) the engine operating conditions including speed, constancy of load and speed, characteristics of lubricants, and the operating temperatures throughout the engine.

Fuel characteristics.—The cetane number of the fuel has an important effect on engine performance. Fuels with low cetane rating have high ignition lag, as is shown by figure 1. A considerable amount of fuel collects in the combustion space before ignition occurs, with the result that high maximum pressures are reached and there is a tendency toward knocking. This tends toward increased wear of the engine, and toward reduced efficiency.

Fuels with high cetane ratings have low auto-ignition temperatures, and therefore are easier starting than fuels with low cetane ratings. These effects are illustrated by figure 2.

Compression ratio.—The thermal efficiency of an engine increases as the compression ratio is increased, as demonstrated in the discussion of cycles in an earlier chapter. The compression ratio of a Diesel engine has a limited minimum value prescribed by the compression required for starting, and is dependent to this extent on the fuel which is used. The upper compression ratio is not limited by the fuel used, but only by the strength of the parts of the engine and by the permissible engine weight per horsepower output.

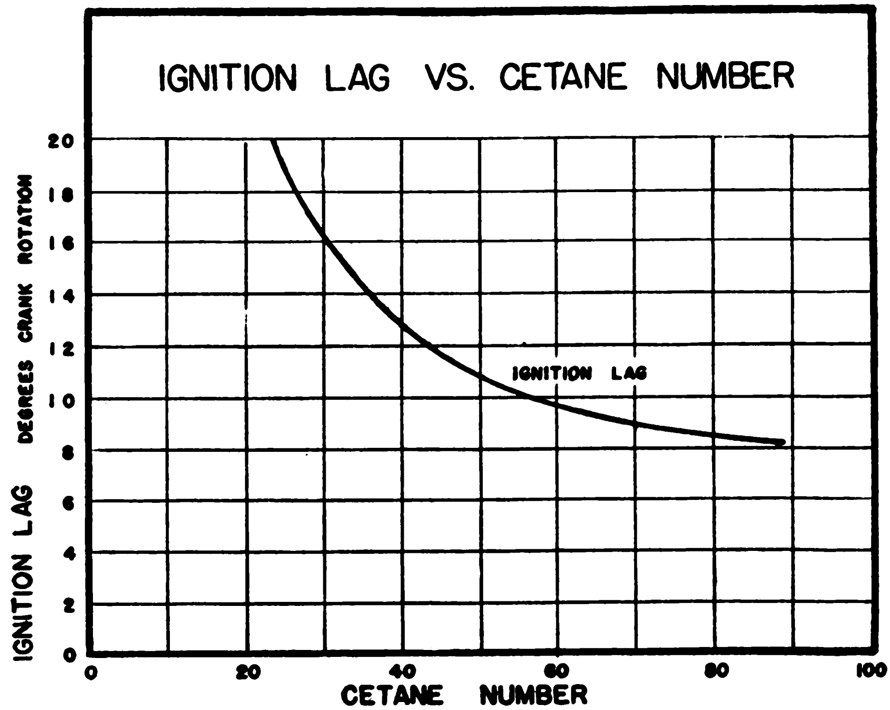


Figure I.

Volumetric efficiency.—The volumetric efficiency of an engine is the ratio of the volume which would be occupied by the air charge at atmospheric temperature and pressure compared with the cylinder displacement (the product of the area of the bore times the stroke of the piston). The volumetric efficiency determines the amount of air which is available for combustion of the fuel, and hence influences the maximum power output of the engine.

Air temperature.—The temperature of the air in the cylinder directly affects the final compression temperature. A high intake temperature results in decreased ignition lag and in easy starting, but is generally undesirable because it decreases the volumetric efficiency of the engine.

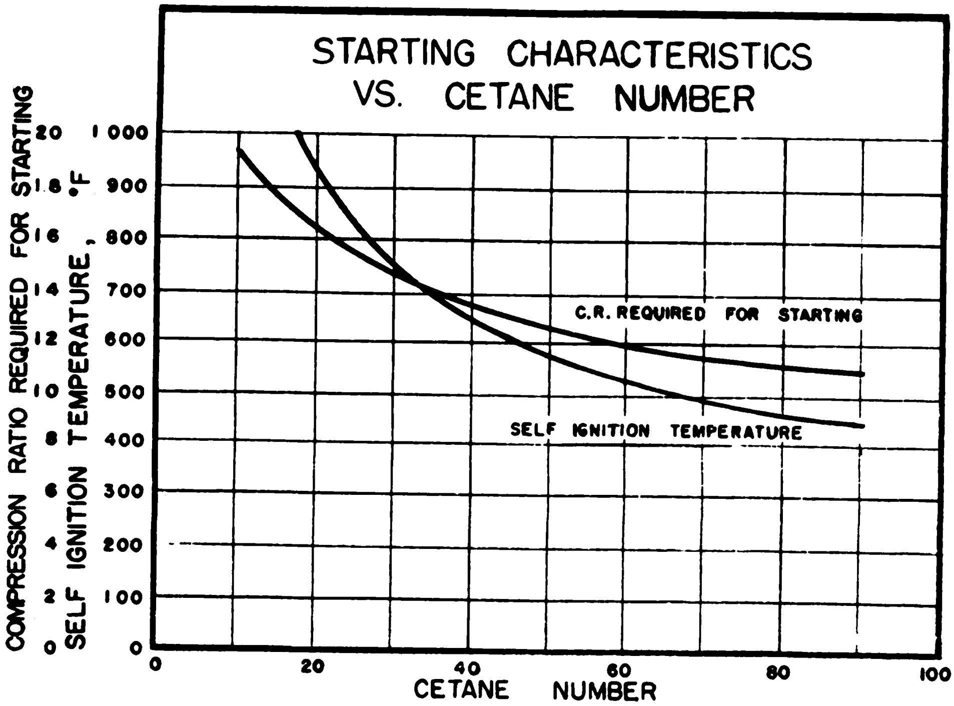


Figure 2.

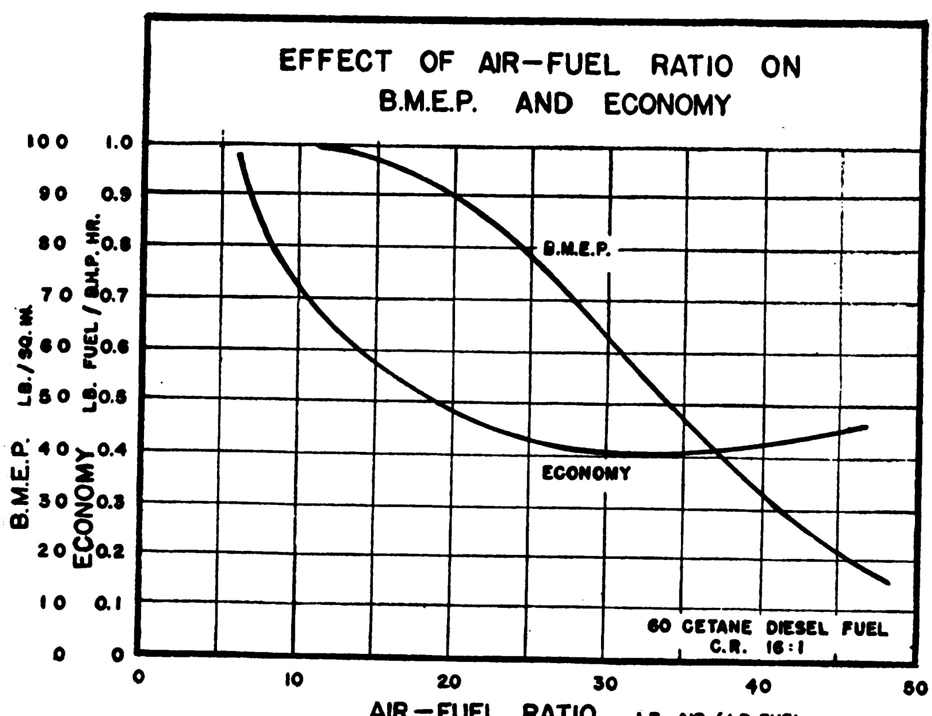


Figure 3.

Quantity of fuel injected per stroke.—The quantity of fuel injected determines the amount of energy available to the engine, and also (for a given volumetric efficiency) the air to fuel ratio. The effect of the air to fuel ratio on b. m. e. p. and economy is shown in figure 3.

Injection timing.—The injection timing has a pronounced effect on engine performance. The optimum is between 5° to 10° before top dead center for many engines, though it varies with engine design. In addition to this effect, early injection tends toward the development of high cylinder pressures because the fuel is injected during a part of the cycle when the piston is moving slowly and combustion is therefore at nearly constant volume. Extreme injection advance will cause knocking. Late injection tends toward decreasing the m. e. p. of the engine and toward lowering the power output. Extremely late injection tends toward incomplete combustion, and the engine operates with a smoky exhaust.

Injection rate.—The rate of injection is important because it determines the rate of combustion and influences the engine efficiency. Injection should start slowly so that a limited amount of fuel will accumulate in the cylinder during the initial ignition lag before combustion commences. It should proceed at such a rate that the maximum rise in cylinder pressure is moderate, but it must introduce the fuel as rapidly as permissible in order to obtain complete combustion and maximum expansion of the combustion products.

Atomization of fuel.—The average size of the fuel particles affects the ignition lag and influences the completeness of combustion. It is desirable to have a small particle size. Opposed to this requirement is the fact that small particles have a low penetration, and there is therefore a tendency toward incomplete mixing of the fuel and the combustion air which leads to incomplete combustion.

Combustion chamber design.—The amount of turbulence present in the combustion chamber of an engine affects the mixing of the fuel and the combustion air. High turbulence is an aid to complete combustion. On the other hand turbulence wastes power and tends to reduce the power output of the engine. Turbulence increases the rate of heat transfer from the gases in the cylinder to the cylinder walls and tends to reduce the compression temperature and to decrease the thermal efficiency of the engine. This must in some cases be offset by partially insulating the combustion chamber from the water-cooled parts of the engine.

STARTING DIESEL ENGINES

Before starting a Diesel engine which has been idle for some time, it must be jacked over slowly with the cylinder cocks open to check against the presence of water in the cylinders. This is particularly important in the case of submarine engines. During the jacking period and at the time of starting, the auxiliary lubricating oil pump must be running if the engine is fitted with it. With the fuel supply shut off, the engine is turned over and brought up to starting speed. Fuel is then supplied to (some of) the cylinders and when the engine begins to carry the load, the starting device is secured and the engine is run at idling speed for a warm-up period.

Small Diesel engines, such as motorboat engines, are cranked by electric starting motors similar to those used for automobiles. Engines used to run direct-current generators, such as submarine engines, are usually started by motoring the engine with the generator. Practically all large engines are started with compressed air.

Compressed air for starting Diesel engines is stored in a pressure flask known as the starting air bottle. For air-injection engines, which are equipped with high-pressure air compressors, the usual starting air pressure is from 500 to 750 pounds per square inch gage. For mechanical injection engines, the usual starting air pressure is from 150 to 300 pounds per square inch gage. Air is led from the starting air bottle through a master valve to a manifold running the length of the engine and is supplied to special starting air valves fitted in at least one-half of the cylinders. These valves are actuated by cams on the camshaft which are provided for the purpose. These valves are arranged to open when the piston reaches top dead center and to remain open from 45° to 90° after top dead center. Air is supplied to all cylinders fitted with starting air valves until starting speed is reached. The starting air is then shut off from half of the engine cylinders and fuel is injected. When these cylinders carry the load of the engine, the master valve is shut, fuel is supplied to all cylinders, and the engine is warmed up.

SPEED CONTROL OF DIESEL ENGINES

The speed of Diesel engines operating within their limit of load capacity is varied by changing the amount of fuel injected into each cylinder per cycle. The speed control of most main drive engines is manual, but a governor is usually provided to prevent the engine from racing when it is suddenly unloaded as, for example, by the propeller lifting out of the water as the ship pitches. Engines used for auxiliary purposes are usually controlled by isochronous governors, and the governor acts to limit the duration and/or rate of the fuel injection.

STOPPING DIESEL ENGINES

Diesel engines are stopped by cutting off the supply of fuel to the cylinders. This is accomplished by adjustment of the fuel pump, rotating the piston to give a zero effective stroke, and interrupting the fuel delivery. An alternate method is to move the camshaft to a neutral position discontinuing operation of the injection valve gear. This latter arrangement is used with air injection, common rail, and unit injector installations, but is not applicable to systems using separate pumps and injection nozzles.

> There have been occasions when Diesel engines, which were running hot and were pumping considerable lubricating oil past the piston rings, have continued to run after the fuel oil was shut off. The engine may be stopped under such conditions by applying a load or by cooling the engine with increased flow of water through the cylinder jackets. Failure of the engine to respond to its controls is very serious. To guard against this possible condition, consumption of lubricating oil must be limited by proper fitting of the piston rings.

Permission must always be secured from the bridge before securing main engines. Usually this permission is given when the bridge signals "Finished with engines".

REVERSING DIESEL ENGINES

The direction of rotation of a Diesel engine is determined by the valve and injection timing. Two sets of cams for operating the intake, exhaust, and starting air valves, and for actuating the injector are provided. One set provides for "ahead" operation and the other set for "astern" operation. Timing diagrams for ahead and astern operation of a four-cycle engine are shown in figure 4.

To reverse a Diesel engine, the engine is stopped and the valve gear is shifted to bring the second set of cams into operating position. The details of the valve gear differ on various types of engines. In general, the ahead and astern cams are mounted on the same camshaft. The reversing gear moves the camshaft fore and aft on some engines to bring one set of cams or the other in line with the rocker arms. On other engines two cam followers are provided for each rocker arm, and the action of the reversing gear is to bring the correct set of cam followers into the operating position.

The necessity of bringing a Diesel engine to a stop before it can be reversed is a severe limitation on direct drive ships. The motion of the ship through the water causes the propeller shaft to continue to turn until the ship has lost most of its way. To overcome this, some ships are fitted with a clutch between the engine and the propeller

shaft which allows the engine to be disconnected for faster reversing.

The engines of submarines are not reversible. To reverse the propeller drive, the main engines are disconnected and the electric drive motors are run astern using battery power.

Special provision for rapid reversing has been included in the installations in recent submarine chasers. These ships are fitted with a fluid drive unit installed between the engine and the reduction gears. This forms a nonrigid coupling in which all power is transmitted from a driving impeller to a driven impeller through the oil which fills the unit. An air brake is fitted on the propeller shaft. When power is cut off the engines, the air brake operates automatically to stop the rotation of the propeller shaft and also, acting through the fluid drive unit, to bring the engine to a stop. The reversing gear is shifted and the engine started in the opposite direction. As soon as the engine speed reaches a predetermined value, the air brake automatically releases and astern rotation of the propeller commences.

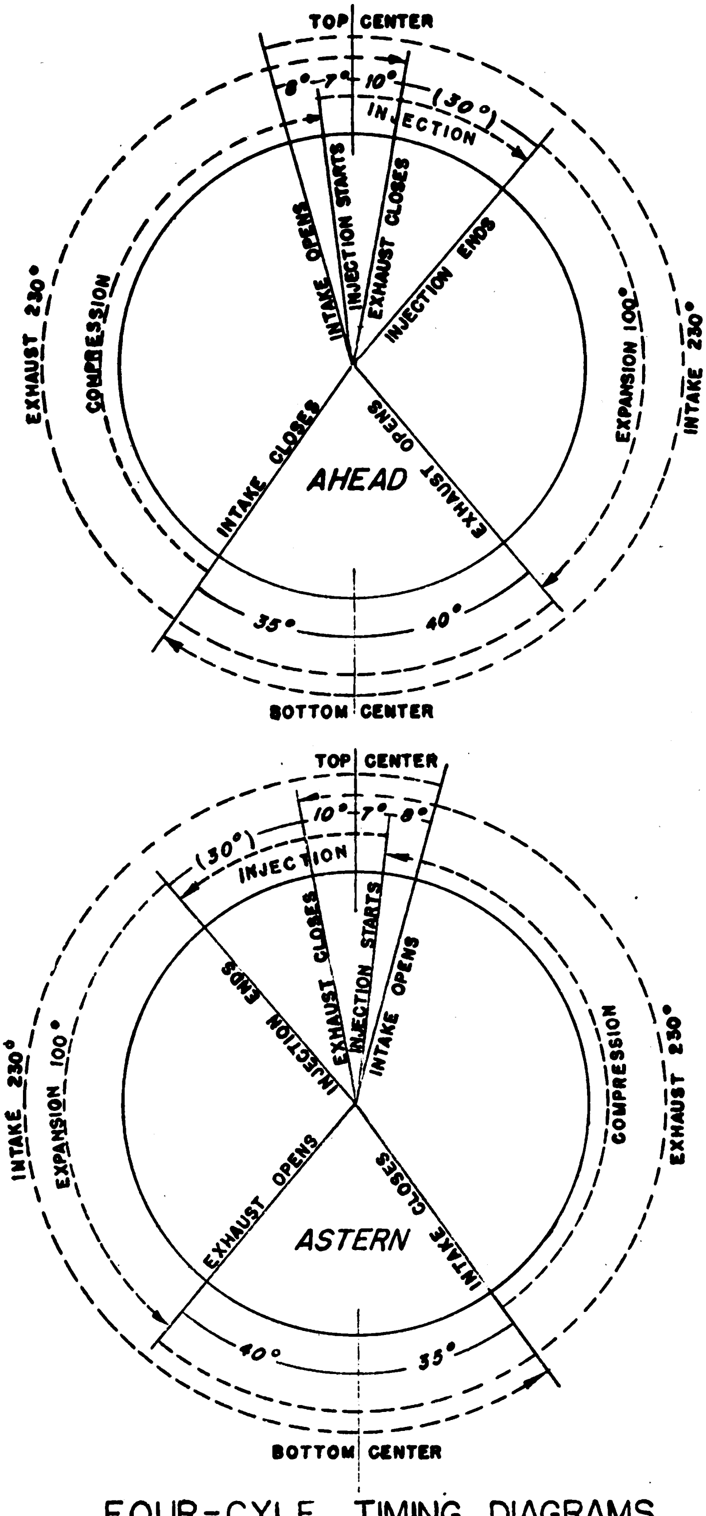
TIMING DIAGRAMS AND ENGINE TIMING

The timing diagrams shown in figure 4 demonstrate the need for two separate sets of cams for reversing a Diesel engine are typical of four-cycle engines. The timing diagram for a typical two-cycle engine is shown in figure 5.

Timing diagrams serve as guides for adjusting and checking the timing of the valves and injectors of engines. The data presented constitutes the manufacturers recommendations for valve settings, and generally summarizes extensive tests conducted with engines of the type for which it has been prepared. It will therefore rarely be expedient to depart from the settings prescribed by the diagram.

The most important setting is that of the injection valve.—Diesel engines are sensitive to exact spray timing. An error of 5 crank degrees is sufficient to cause a considerable decrease from optimum engine performance. Settings should be made to *within 1 crank degree* in the manner prescribed by the operation manual for the engine.

Valve timing is less critical than injection timing. A variation of 5 crank degrees from the specified values is permissible, though not desirable. Modern engines usually are not provided with adjustable valve cams. Normally, no adjustments will be required other than to maintain the specified clearances between the rocker arms and the valve stems while the valves are in the closed position. This adjustment is very important. Typical clearances with the engine cold are 0.031 inch for air intake valves, 0.035 inch for exhaust valves, and 0.035 inch for air starting valves. Failure to maintain enough clearance may prevent the valves from closing completely when the engine

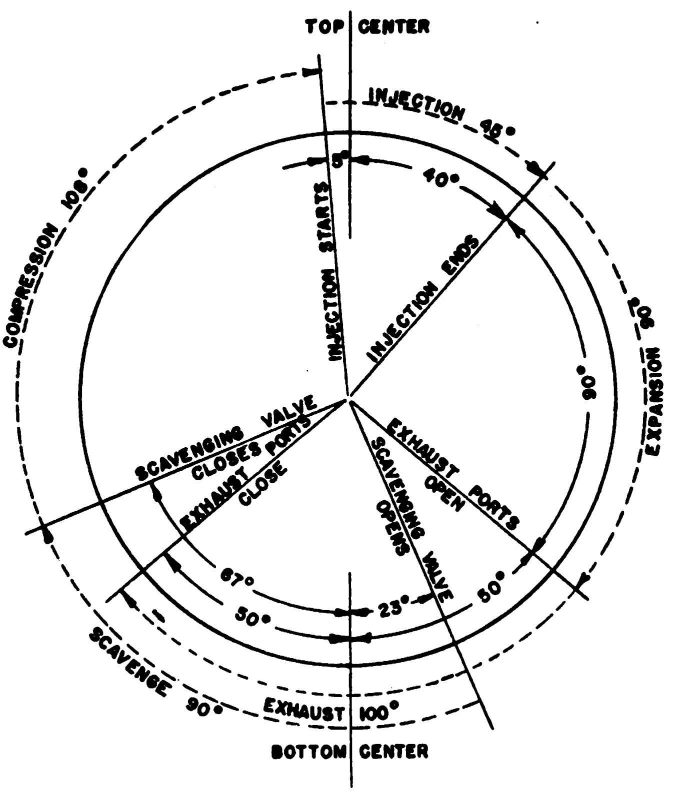


FOUR-CYLE TIMING DIAGRAMS

Figure 4.

is hot. This causes rapid erosion of the valves and valve seats, and loss of compression. Too great a clearance will subject the valve to pounding in opening and closing, and will decrease the time that the valve is open during the engine cycle.

The cycle followed by each piston may be traced around a timing diagram. For instance, referring to figure 5, starting with the piston on the up stroke it is seen that injection starts 5 crank angle degrees before top dead center and continues for 45°, ending 40° past top dead



TWO-CYCLE TIMING DIAGRAM

Figure 5.

center. At this point combustion is perhaps three-quarters complete. The piston, now on the down stroke, exposes the exhaust ports 50° before bottom dead center, and 27° later, or 23° before bottom dead center, it exposes the air intake or scavenging ports. Release of burnt gases starts when exhaust ports open, and expulsion of combustion products is greatly hastened by the sweeping action of the fresh air charge which starts rushing in when the scavenging ports are exposed. The exhaust ports stay open 100°, closing 50° past bottom dead center. 17° later, or 67° past bottom dead center, the scavenging ports close.

Thus, quick and thorough scavenging of exhaust gases has been

obtained by a wide "overlap," 73° in this case. After the scavenging ports have been closed, the fresh air which has entered the cylinder is compressed as the piston moves up again. At 5° before top dead center, injection again occurs, and the cycle repeats. This timing diagram apparently applies to a medium-speed engine.

It is noticed that all cyclical events occur within 360 crank degrees, or one engine revolution, in the *two cycle engine.* The particular engine represented in figure 5 has *ports;* however, the same timing could be obtained with an engine equipped with intake and exhaust valves. (See fig. 3, Fundamentals, II). The timing of ports, of course, is automatically done by the piston moving up and down in its cylinder. However, *valve* and injection timing must be set externally by timing the camshaft, and adjusting tappet clearances.

A four-cycle engine timing diagram can be traced in the same manner. Two revolutions of the diagram must be made to include all events, since the four-stroke cycle covers 720 crank degrees. Referring to figure 4 for AHEAD rotational direction, the piston coming up to top dead center has compressed an air charge taken in on a previous cycle. Injection starts at 7° before top dead center and ends 40° past top dead center. Combustion starts within the injection period, and ends shortly thereafter. The piston then is moving down on its expansion (or working) stroke to a point 40° before bottom dead center, when the exhaust valve opens and releases the burnt gases. Residual pressure forces some gases out before the piston reaches bottom dead center, and after that point the piston forces out the remainder of the burnt gases as it moves up toward top dead center. At a point 15° before top dead center, the intake valve opens and fresh air starts in (forced by the supercharger, if installed). The exhaust valve is still open and does not close until 25° later, or 10° *past* top dead center. Thus a 25° valve overlap is obtained, which aids in scavenging and reduces the fresh-air dilution by exhaust gases which might otherwise remain in the clearance volume. From here the piston continues to draw in fresh air as it moves down. The intake valve closes at 35° *past* bottom dead center. The intake valve seldom is set to close right at bottom dead center, since a short delay beyond that point not only does not cause air to flow back out (remember, the piston moves slowest near dead center points), but actually allows more air to flow in due to a "ramming" action; that is, air flowing in through the air induction system possesses some inertia, and will continue to flow momentarily after the motivating force (piston suction) has ceased. Supercharging, of course, accentuates this effect. After the intake valve closes, the air charge is compressed as the piston moves upward toward the point where injection will occur as another cycle begins.

FINDING TOP DEAD CENTER

In order to time an engine, it is sometimes necessary to place a piston accurately at its top dead center position. Usually, the engine flywheel will be plainly marked relative to reference mark on the frame to locate top dead center. However, if such marking is not present, or if present but believed inaccurate, top dead center can be found by the method given hereafter.

Expose the pistons by removing the cylinder head. Bar the engine over until the piston in number one cylinder (cylinders always number from forward to after or driving end of the engine) moves up to a point roughly an inch from its point of highest travel (top dead center). Mark the flywheel against any convenient *fixed* point on the frame adjacent to the flywheel. Install a dial indicator gage on a firm base over the piston in such position that an extension rod screwed into the spindle of the dial rests on any convenient point on the working face of the piston. Care must be exercised to place the rod at a point where it will not slip as the piston moves. Set the dial at zero on its scale. Now, crank the engine in the *same direction* as before, until the piston has moved up to top dead center and back down to the same point, as indicated by the dial which will again read zero. Mark this position on the flywheel.

There are now two marks on the flywheel, each equi-distant from the top dead-center point between them. Using a flexible steel rule, or equivalent, measure the circumferential distance between points. Divide this distance in half, and lay it out along the circumference of the flywheel from one of the original marks. Check this position by measuring from the other point. The point thus obtained by bisecting the trial crank angle marks the top dead-center point. When the engine is cranked (always in the same direction, to avoid "backlash," i. e., "play" in engine parts) until the flywheel top dead-center point is just at the reference point on the frame, the number one piston will be at its top dead-center position.

The procedure should be repeated several times to verify first results. After the top dead-center point is located accurately, it should be permanently marked with a prick punch, for future use.

The top dead-center position of any other piston can be located on the flywheel by laying out on the flywheel the angle by which the particular piston is out of phase from number one piston. However, all timing is usually done using only one cylinder, since the cams on the camshaft are fixed, and will fall in proper cyclical position if any one of them is properly set.

TIMING OF VALVES

The universal method of actuating valves is by cam action. Modern camshafts have cams which are integral with the shaft, and hence

cannot be moved relative to one another without deforming the entire shaft. Valve timing simply means locating the camshaft relative to the followers, push rods, and tappets so that the cam will start lifting the valve at the correct instant in the cycle.

Timing of valves is accomplished by meshing the timing gear (on the drive end of the camshaft) with its mating gear in the gear train, in their proper relative position. The teeth on the two gears which should mesh are usually marked by the manufacturer. Sometimes, though, the gears are not so marked; or, occasion may arise wherein it is desirable to change the valve timing somewhat. In such cases, the piston should be brought to the point in the cycle where the particular valve being timed should *start to open.* The timing gears should then be meshed in such position that the cam on the camshaft which corresponds to the valve being timed *just starts* to opén the valve (through the follower, push rod, rocker arm, and tappet). The piston can be located in the proper position by laying out the desired angle on the flywheel from the top dead-center point, and cranking the engine to this position. The angle to be laid out can be picked off the engine's timing diagram.

Somewhere in the cam follower—push rod—rocker arm—tappet system provision is always made for adjusting tappet clearances. These clearances should be set exactly as recommended by the manufacturer of the engine.

CLASSIFICATION OF DIESEL ENGINES

Diesel engines may be classed (1) according to their operating cycle as *two-cycle* and *four-cycle* engines, (2) by major mechanical features (single or double action, cylinder arrangement, etc.), (3) by the method of fuel injection as *air-injection* or *solid-injection* engines, and (4) by the type of combustion chamber as open *combustion chamber*, *precombustion chamber*, *turbulence chamber*, and *air cell* engine.

Classifications (1) and (2) have been considered in the chapters entitled "Engine Fundamentals, 1," and "Engine Fundamentals, 2". Classifications (3) and (4) will be considered in the following chapters on Diesel engine.

2. AIR-INJECTION SYSTEMS

THE PRINCIPLES OF AIR INJECTION

The function of any Diesel fuel injection system is to deliver fuel to the engine cylinders. As previously set forth, the requirements for a satisfactory fuel injection system demand (1) that it delivers an exactly metered amount of fuel to each cylinder, supplying the quantity required for the load being carried by the engine, (2) that it delivers this fuel on a definite time schedule, commencing at the

correct point in the cycle and progressing at a specified rate, and (3) that it injects the fuel in a finely divided form with sufficient velocity for adequate penetration into the highly compressed combustion air in the engine cylinder.

The first system developed which adequately met these requirements was the air-injection system. Air injection was utilized by the first successful Diesel engines (in about 1900) and was universally employed until about 1930. The air injection system has now been superseded, and since 1935 very few engines have been constructed with air-injection fuel systems. There are, however, a large number of air-injection engines still in use at the present time (1942).

A schematic diagram of an elemental air-injection system is shown in figure 1. The system is composed of three distinct elements: (1) an air compressor for compressing the injection air, (2) a fuel pump for metering the fuel supplied for each power stroke of the engine, and (3) an injection valve for atomizing the fuel and controlling the injection rate and timing.

Compressed air, at a pressure of 750 to 1,000 pounds per square inch gage, is supplied to the air-injection nozzle. This pressure greatly exceeds the compression pressure in the engine cylinder, the latter reaching a pressure near 500 pounds per square inch gage. Fuel oil is delivered to the nozzle by the fuel pump, which is a positive displacement pump so constructed that the amount of fuel delivered per stroke of the pump may be varied to suit the engine load requirements. At the proper instant in the cycle, determined by a cam on the engine camshaft, a needle valve in the injection nozzle is opened, and a stream of air from the nozzle enters the combustion chamber at a very high velocity, atomizing the fuel charge and projecting it into the compressed air in the combustion space.

While the injection air is available for the combustion of fuel in the engine cylinder, its primary function is to project the fuel into the combustion space. The principal portion of the air used for combustion is supplied to the engine at or near atmospheric pressure, and enters the working cylinders from the intake manifold through the intake valves.

INJECTION AIR COMPRESSORS

The compressor in figure 1 is typical of those used with air-injection systems. It is a three-stage compressor; that is, it compresses the air in three steps, or stages, from atmospheric pressure up to the injection pressure. Between stages, the compressed air is cooled in the intercoolers, coolers 1 and 2. This is done for two related reasons. The air is heated as it is compressed to a degree dependent upon the per-

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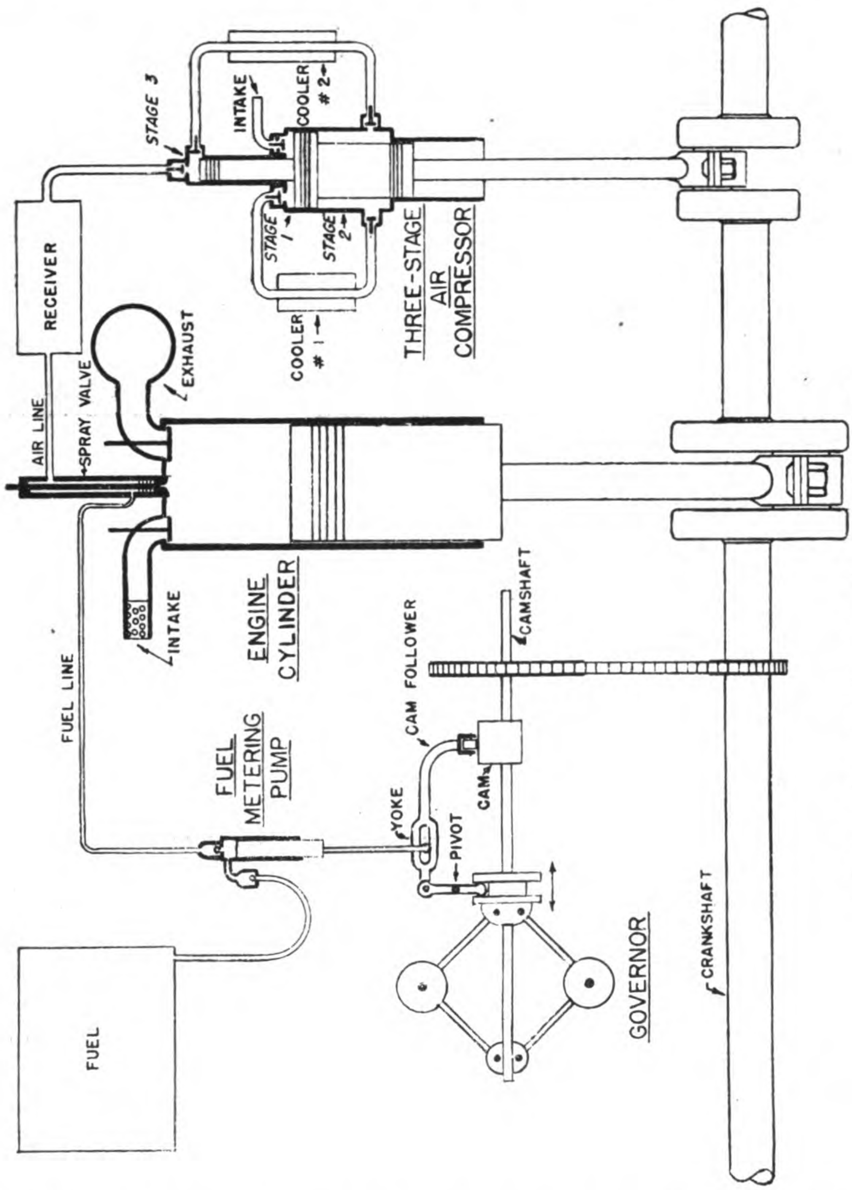
centage increase in pressure, and as the result of this it occupies a greater volume than it would at the same pressure but at its original temperature. Since the amount of work required for a definite increase in pressure is dependent on the volume of air compressed, a



AIR-INJECTION DIESEL ENGINE

Figure 1.

considerable saving in work is obtained by cooling the air between stages. Further, by the use of intercoolers, the final temperature of the air leaving the compressor is kept below a safe limit.

On each down-stroke, the compressor takes a full charge of air through the intake connection. On the following up stroke it is compressed in the annular space marked "Stage 1" and is pumped at moderate pressure through cooler 1 into the space marked "Stage 2."

On the second down stroke, the moderately compressed air in the second stage is compressed to a higher pressure and is delivered through cooler 2 to the space marked "Stage 3." In Stage 3, during the second up stroke, it is finally compressed and is pumped to the receiver at the injection pressure. In some installations a final cooler is provided. In others, as in figure 1, the pipe between the compressor and the receiver serves as the final cooler.

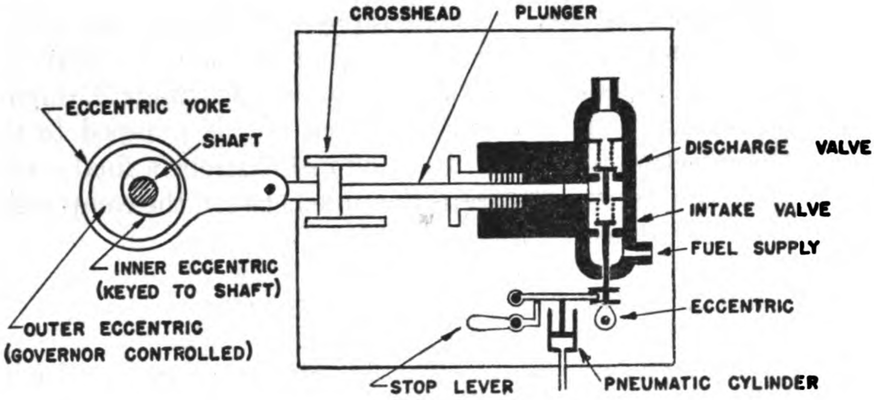
METERING PUMPS FOR AIR-INJECTION SYSTEMS

There are three general types of variable delivery pumps used with air-injection systems constructed prior to about 1930–35. The methods of delivery control used are: (1) variation of the pump plunger stroke, (2) variation of the time for closing of the intake valve of the pump in relation to the stroke of the pump plunger, and (3) variation of the time of closing of a bypass valve which returns to the suction line a part of the oil handled by the pump.

Variable stroke pump.—The piston motion of the usual forms of variable stroke pumps is obtained through the use of an eccentric operated by the camshaft. A linkage, adjusted manually or by the governor action, alters the effective throw of the eccentric. A schematic linkage is shown in figure 1 for purposes of illustration. The governor acts so that an increase in engine speed moves the governor spool to the left and in consequence the link, rotating about its fixed pivot point (solid black in the illustration), moves the cam follower to the right so that the rod from the pump plunger engages the yoke closer to the pivot of the cam follower. A smaller fraction of the cam motion is then transmitted to the pump, decreasing the amount of fuel delivered and thus tending to reduce the speed of the engine.

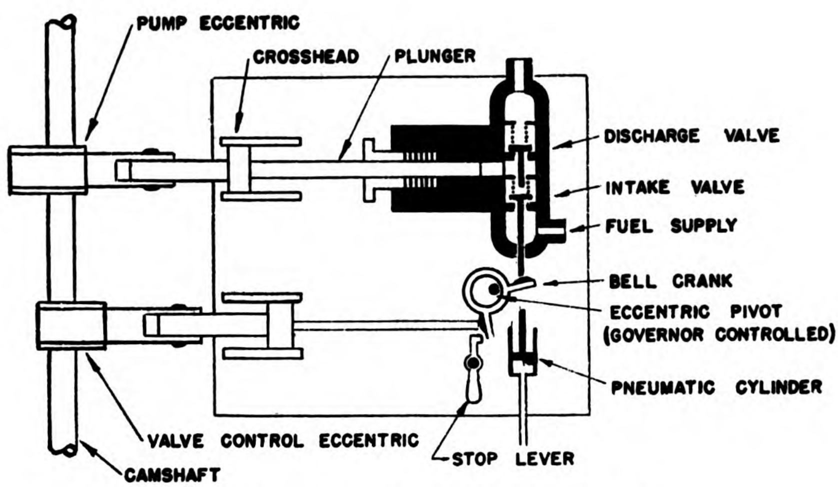
The essential parts of the linkage actually used in a commercial pump of McIntosh & Seymour construction are illustrated in figure 2. The pump shown in figure 2 is governor controlled. An eccentric is keyed directly to the camshaft, and on this inner eccentric a second eccentric is mounted. The governor (which is not shown) is also mounted on the camshaft and operates through a linkage to turn the outer eccentric on the inner one. The effective eccentric throw is the vectorial sum of the throws of the individual cams, and may be altered between zero and a maximum value.

The basic advantage of this arrangement is that very little load is imposed on the governor to make and maintain a change in the setting. The eccentric yoke follows the motion of the combined eccentrics and transmits a reciprocating component of the motion to the pump piston, which is steadied by a crosshead. Fuel oil is supplied to the pump from a gravity feed tank through a filter. In order to obtain positive valve action in spite of low pressure on the supply line, the



VARIABLE PLUNGER STROKE DIESEL FUEL PUMP

Figure 2.



SUCTION VALVE CONTROLLED DIESEL FUEL PUMP

Figure 3.

pump suction valve is opened against the force of the suction valve spring by a cam. The outlet valve is spring loaded and lifts when the pump delivery pressure exceeds the pressure in the outgoing fuel line. A hand lever is provided to open both pump valves when it is necessary to stop the engine.

Pumps with variable suction valve timing.—The pump shown in figure 3 utilizes a variation of the time of closing the intake valve of

the pump as a means of delivery control. Two eccentrics are keyed to the camshaft. One of these eccentrics provides a constant reciprocating motion for the fuel pump plunger. The second eccentric provides a constant reciprocating motion to a tappet which acts through a bell crank to control the closing of the suction valve of the pump. The bell crank is mounted on an eccentric pivot. Rotating this pivot changes the effective arm ratio of the bell crank, resulting in a change of the timing of the suction valve closing with relation to the pump plunger stroke. The engine governor is linked to the eccentric pivot of the bell crank, and thus has control of the amount of fuel delivered to the engine.

A small pneumatic cylinder is provided which opens the suction valve of the pump when air for starting the engine is being supplied to the engine cylinder the pump serves. This prevents injection of fuel before the engine is up to speed.

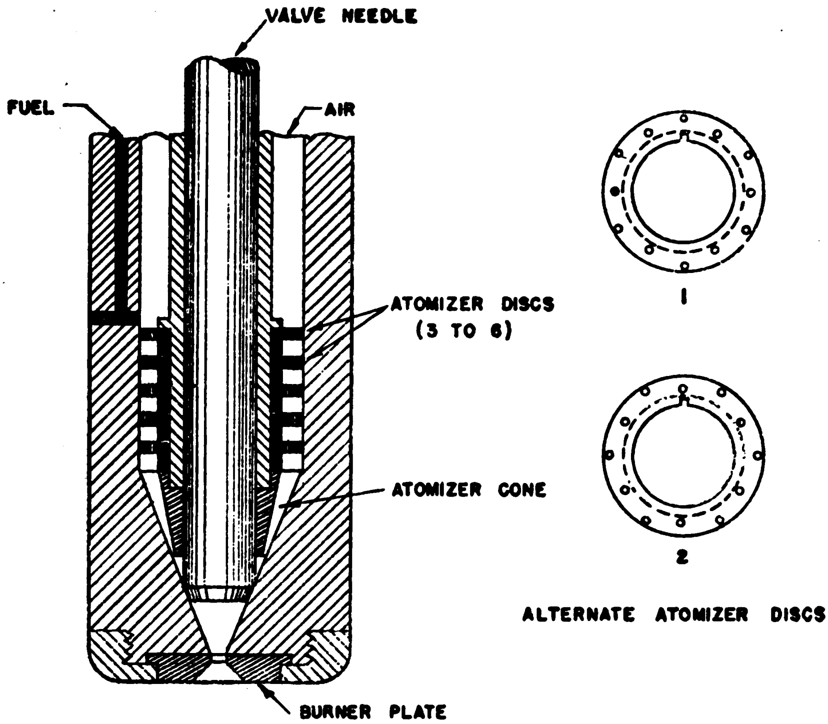
Variable bypass pumps.—The linkage required for pumps with bypass control is similar to that used with variable suction valve timing. Variable bypass pumps are uncommon, and no illustration is included in this text.

Recent practice.—Engines constructed since 1935 have frequently employed Bosch pumps of the type to be described later under the heading of "solid injection systems."

AIR INJECTION FUEL NOZZLES

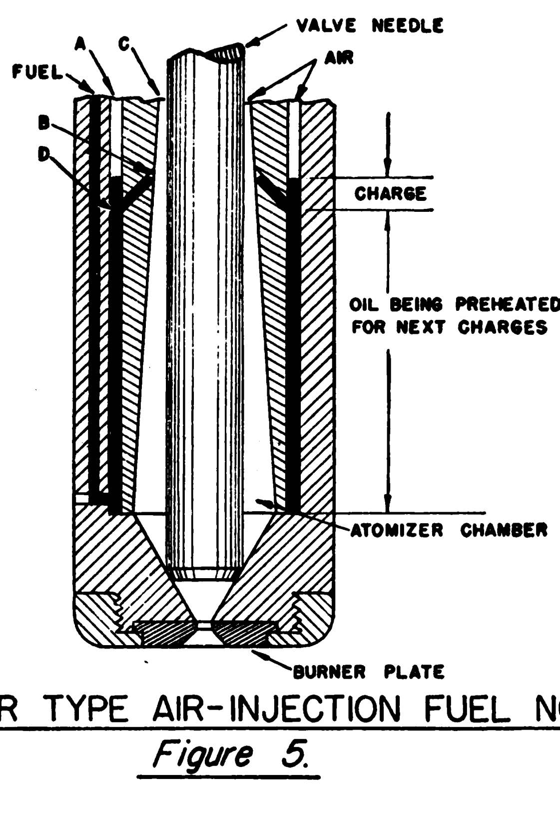
There are two principal types of air injection fuel nozzles. In the atomizer disk type, the fuel and air pass together through a series of perforated disks which break up the oil into small droplets and form an air and fuel emulsion. This is injected into the engine cylinder through the burner plate orifice (or orifices). In the jet type a high velocity stream of air aspirates fuel from the supply chamber and tears off tiny droplets which are further broken up and injected with the air through the burner plate into the engine cylinder.

Atomizer disk nozzles.—The construction of a typical atomizer disk-type nozzle is shown in figure 4. Fuel is delivered from the metering pump while the needle valve of the nozzle is closed and is deposited on top of the assembly of atomizer disks. When the nozzle needle valve is opened, the pressure of the injection air forces the fuel through the holes in the atomizer disks. The holes in alternate disks are not in line, and the jets of fuel sprayed through the holes are broken into droplets by impinging on the disk below, being thoroughly mixed with the injection air. Each succeeding disk breaks up the droplets further and produces a fine fog of oil droplets mixed with injection air. This fog passes through nozzle-shaped passages in the atomizer cone below the bottom disk where it is further



ATOMIZER DISC TYPE AIR-INJECTION FUEL NOZZLE

Figure 4.



JET ATOMIZER TYPE AIR-INJECTION FUEL NOZZLE

Figure 5.

mixed, and finally the emulsion is sprayed through the orifice in the burner plate into the engine cylinder.

An injection air pressure of about 900 pounds per square inch gage is required when the engine is operating at full speed. At slow speeds, since the valve is open for the same number of crank degrees each cycle and hence for a longer actual time each cycle than when the engine is operating at high speeds, it is necessary to reduce the injection air pressure to prevent blowing an excessive amount of injection air into the cylinder after all of the fuel is injected. The air pressure may be reduced to as little as 600 pounds per square inch gage, but must *never* be less than abopt 50 pounds per square inch in excess of the compression pressure reached in the engine cylinder.

After the needle valve of the injection nozzle is closed, some of the fuel remaining in the unit drains to the bottom of the unit and collects at the seat of the needle. This part of the charge is injected appreciably ahead of the main charge on the next cycle and has time to ignite before the main charge enters the cylinder. This assists in the smooth operation of the engine and reduces the probability of knocking.

A part of the fuel charge deposited on top of the upper atomizer disk drains through the holes in this disk to the second disk. The proportion which does this is dependent upon the viscosity and the surface tension of the fuel. The amount of atomizing effect varies as the fuel is blown through more or fewer disks during injection. A heavy fuel oil, because it stays on the upper disk, receives the maximum effect. Light fuel oil receives less of the atomizing effect, but breaks up more easily. Consequently, this type of nozzle is capable of handling a considerable variety of fuels and will produce about the same degree of atomization for all grades.

Jet atomizer nozzles.—Figure 5 shows a common construction for a jet atomizer fuel nozzle. The fuel oil enters an annular space in the body of the nozzle which has enough capacity for several fuel charges at full load operation. The fuel is heated in this space. The fuel to be injected is furnished by the fuel pump and displaces an equal amount of heated oil into the reservoir space above the base of the sloping holes (marked D on the diagram). The full injection air pressure acts on the open top of the reservoir space (at A on fig. 5). When the valve needle is lifted, injection air rushes at high velocity along the needle (at C) into an expanding passage. The resultant pressure drop causes oil to be forced from the reservoir up the sloping holes by the differential pressure developed between A and B. From the ends of these holes (marked "B" on the diagram) the oil is torn into tiny droplets by the high-velocity air stream. The turbulence of the air in the atomizer chamber thoroughly mixes the air and oil into a

fog, and this is injected into the engine cylinder through the burner plate.

As in the case of the atomizer disk type of nozzle, some oil collects at the needle seat after the valve closes and this acts as an ignition charge. The preheating chamber enables this type of nozzle to handle heavy fuels.

The air pressure required for jet type atomizers is about the same at a given speed as that required for atomizer disk type nozzles.

Care of nozzles.—It is necessary that nozzles be kept in good condition. Failure of the nozzle to work properly will cause the engine to run erratically.

Particular care must be taken to prevent the nozzles from sticking or clogging as the result of gum formation from thermal decomposition of the fuel oil. Clogged atomizers or burner plates lead to smoky exhaust and are usually at fault when this occurs at moderate loads. It may sometimes be necessary when burning heavy fuel oil to run the engine with kerosene for a few minutes each watch to flush gum from the nozzles. With distillate fuels this is not often necessary. Kerosene should be injected around the valve stem about once in 25 hours to prevent the needle from sticking.

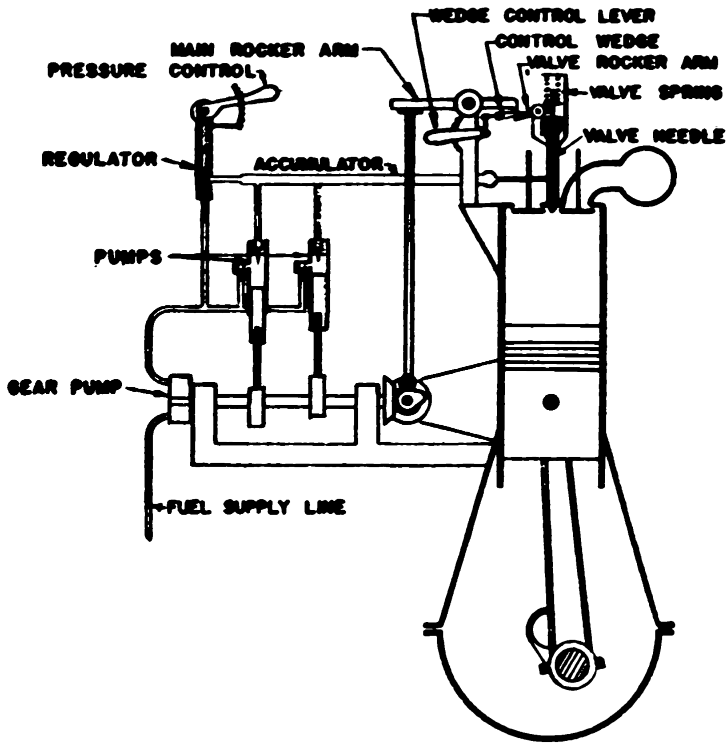
The valve needle must seat firmly along a line contact (though it is not essential that it touch over its entire surface). If the needle does not seat properly, there is danger of hot combustion gases blowing back into the nozzle, burning the holes in the burner plate and the valve stem and seat. Leaky valves sometimes drip fuel into the engine cylinder during the compression stroke of the engine and may lead to violent preignition. Valves must usually be reseated after 2,000 to 4,000 hours of operation.

The valve spring tension must be maintained. Either too little or too much pressure will cause chatter in closing and will result in excessive wear of the needle valve seat.

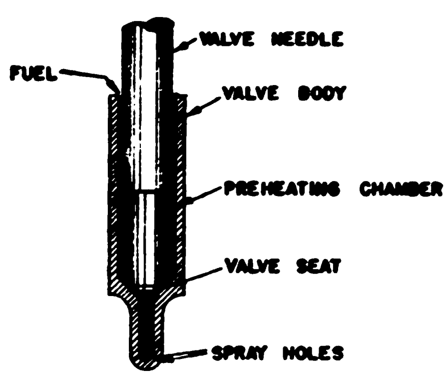
DISADVANTAGES OF AIR-INJECTION SYSTEMS

Air-injection fuel systems are complicated, heavy, and wasteful of power. All of the energy which heats the injection air during compression is wasted, and since the high-pressure air is expanded irreversibly from the injection air main pressure into the combustion chamber, all of the work required to compress the injection air above the compression pressure reached in the working cylinder is also wasted. The total power lost is between 7 and 15 percent of the power developed by the working cylinders of the engine.

In addition to the power losses, the expansion of air through the injection nozzle cools the air and chills the air in the combustion space, contributing to the injection lag and to the difficulty of starting the engine.



DIESEL ENGINE WITH CAM-OPERATED VALVE
COMMON RAIL SOLID INJECTION SYSTEM
Figure 6.



DETAIL OF CAM-OPERATED VALVE
Figure 7.

3. COMMON RAIL INJECTION SYSTEMS

To avoid the disadvantages of air-injection systems, airless or so-called solid injection systems have been developed. In systems of the latter type, the oil is compressed to a very high pressure and the sudden reduction in pressure as the fuel is injected into the cylinder from the injection nozzle results in atomization of the fuel as it expands.

The first successful solid injection systems were of the common rail type. Figure 6 shows the elements of the system. The fuel is compressed to a high pressure by one or more piston pumps, having together somewhat more capacity of fuel delivery than is required by the engine for fuel power operation. A pressure of 2,000 to 8,000 pounds per square inch is maintained in an accumulator, which usually takes the form of a large manifold with a capacity of a large number of injections, and the excess oil pumped is bypassed back to the pump suction line by a pressure regulating valve.

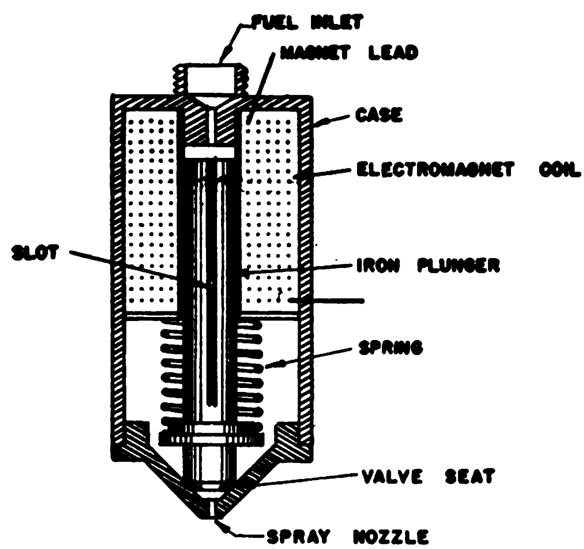
Each cylinder of the engine is fitted with an injection valve connected to this manifold. The valve shown in figure 6 is mechanically operated from the engine camshaft, and the length of time it is held open during each cycle is controlled by governor action.

A detail of a cam-operated spray valve is shown in figure 7. The valve is somewhat similar to an air injection valve, but is much simpler. The flow is controlled by a spring closed needle valve. The fuel is injected into the engine through one or more spray holes in the tip of the nozzle. Absolute tightness of the valve closure is essential because full pressure is exerted on the fuel supply at all times.

The cam operated spray valves are subject to the same change in absolute length of injection time as is experienced with air injection engines when the engine speed is varied. To compensate for this fault, the setting of the pressure regulating valve is altered so that a low pressure is maintained in the accumulator while the engine is running slowly, and a higher pressure is maintained during fast operation.

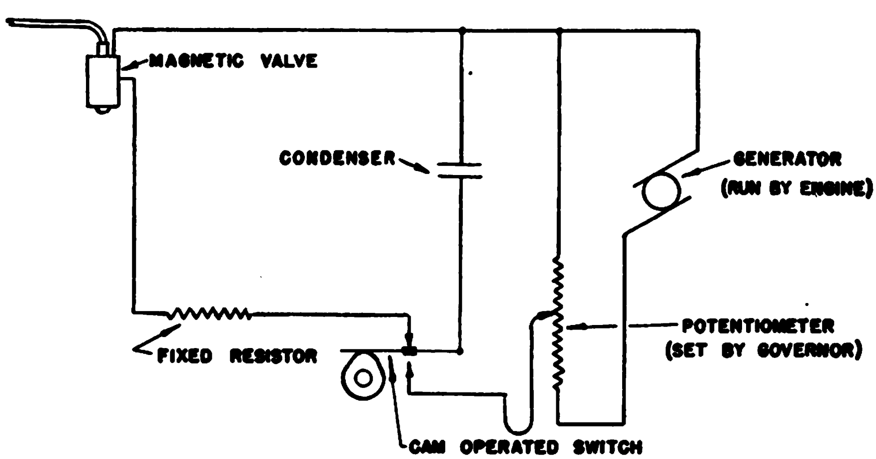
Mechanically operated spray valve common rail systems are rarely constructed at the present time. Recently, magnetic valves for common rail injection systems have been developed and seem to offer promise of wide use because they are inexpensive to construct and may be very accurately controlled, with the result that some saving in fuel consumption is realized for a given power output. Engines fitted with magnetic valves may be maintained at very constant speed regardless of the power-load fluctuations imposed on the engine.

A section through a magnetic spray valve is shown in figure 8. The valve is spring closed and opens when the magnet coil is energized. The electrical circuit used is shown in figure 9. Current is generated at a voltage determined by the engine speed, and is stored in a condenser at a voltage controlled by a potentiometer actuated by the governor. The amount of energy stored in the condenser is dependent on the charging voltage. At the correct point in the engine cycle, the condenser is disconnected from the charging circuit and connected to the magnet circuit. Passage of current through the magnet circuit opens the spray valve and holds it open for the time required



SCHEMATIC SECTION OF MAGNETIC VALVE

Figure 8.



ELECTRICAL CIRCUIT FOR MAGNETIC VALVE
COMMON RAIL SOLID INJECTION SYSTEM

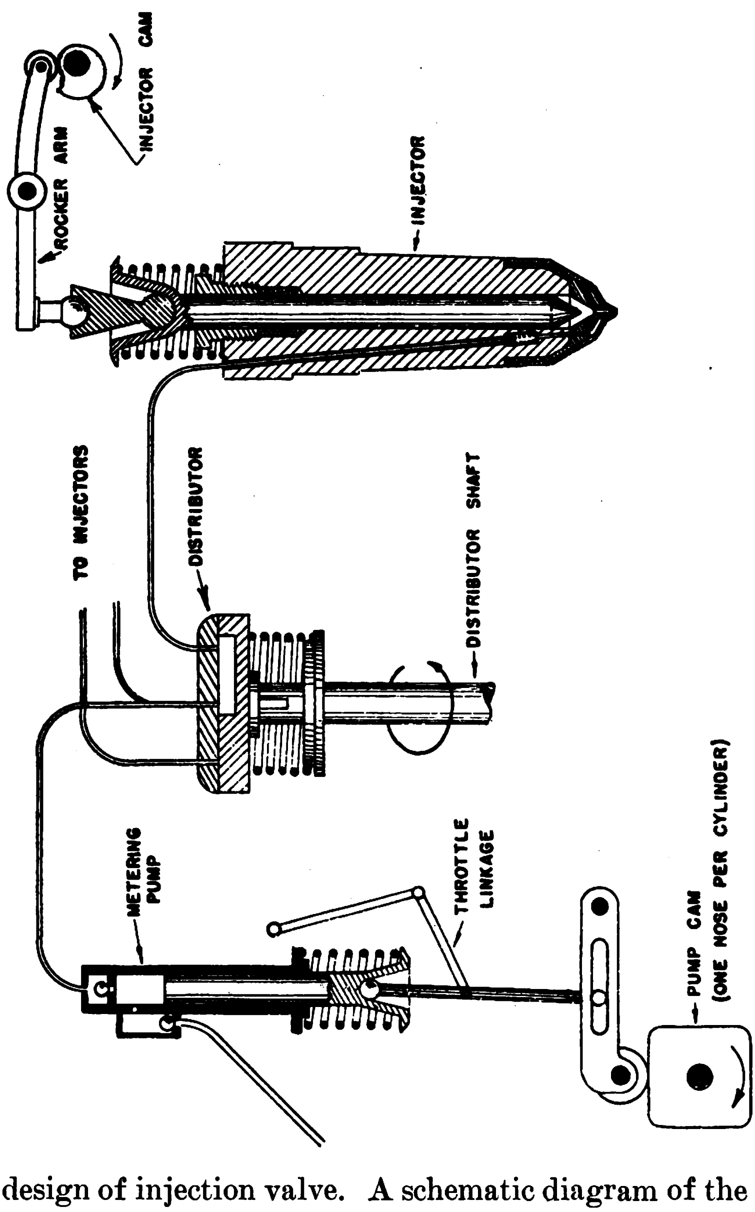
Figure 9.

to discharge the condenser. Operation of the system is precise because the amount of energy stored in the condenser and the length of time required to discharge the condenser are exactly reproducable.

4. SOLID-INJECTION SYSTEMS

THE CUMMINS INJECTION SYSTEM

The Cummins injection system cannot properly be classified as either an air-injection system or a solid-injection system. The distinctive features of the system are (1) the use of a single variable-stroke low-pressure fuel-metering pump with a distributor, and (2) a unique



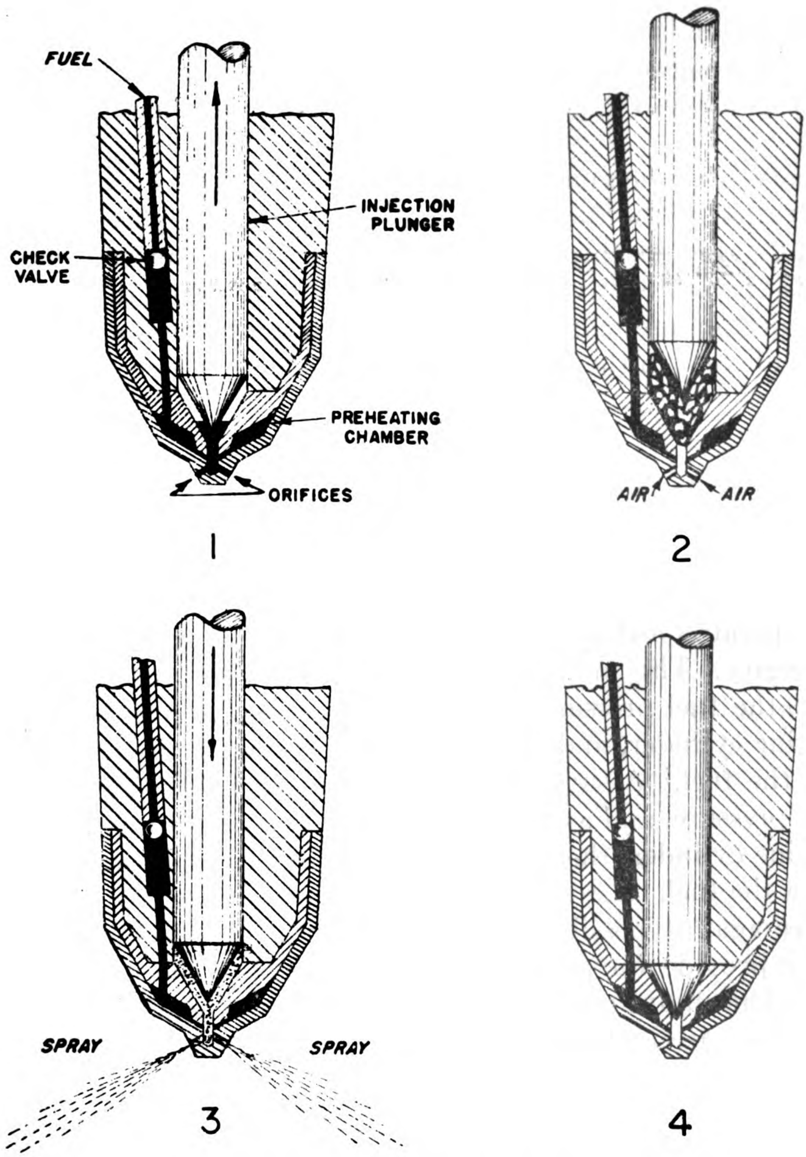
CUMMINS INJECTION SYSTEM

Figure 1.

design of injection valve. A schematic diagram of the system is shown in figure 1.

Fuel is drawn from the feed tank and is supplied at constant pressure to the suction valve of the fuel-metering pump by a gear pump. The gear pump handles an excess of fuel over the requirements of

the engine, and the surplus is bypassed back to the suction line through a relief valve. These parts are not shown in figure 1. The metering pump is of the variable-stroke type. It delivers fuel at low pressure to the injection valve selected by the distributor. The distributor is a rotary valve which is spring loaded so that if excessive pressure is built up by faulty operation of the system, the lower half of the distributor valve will be forced downward, relieving the pressure on the system.



OPERATION OF CUMMINS NOZZLE

Figure 2.

A detail of the Cummins nozzle and a series of four stages during the operating cycle of the engine are shown in figure 2. (1) During the intake stroke of the engine, the valve plunger is allowed to lift by the action of the injector cam, and the metered charge of fuel is delivered to the nozzle. (2) On the following engine stroke, hot compressed air from the engine is forced through the spray orifices from the engine cylinder, heating the fuel and forming a foam of air in the fuel oil. (3) Near the end of the compression stroke of the engine, the injection plunger is forced down by the action of the injection cam and linkage, and the air and oil mixture is sprayed into the combustion space of the engine cylinder. (4) At the completion of injection, the plunger dwells in the down position until the cycle is repeated by the engine.

The advantages claimed for the system over those of solid injection are operation of the fuel-metering pump at low pressure, equal delivery to all cylinders through the use of the distributor, and avoidance of elastic pressure waves in the fuel lines. Some of the advantages of air injection are gained without the serious power waste and great weight inherent in the air injection system.

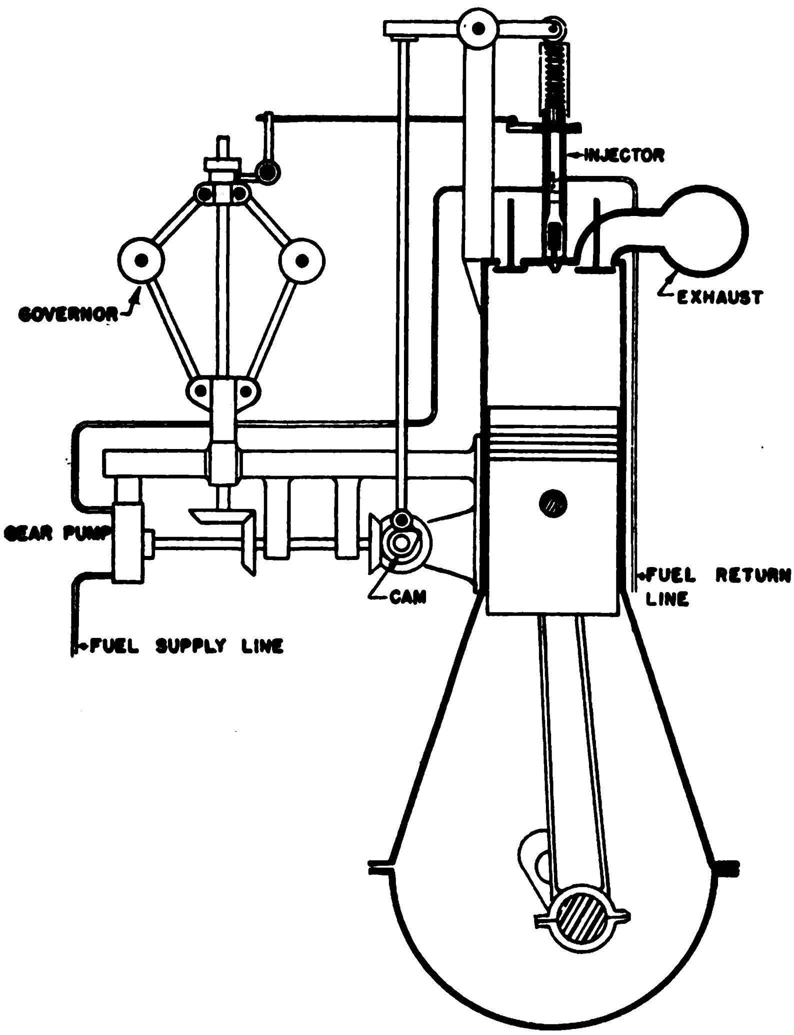
JERK PUMP SYSTEMS

There are two major classes of solid-injection systems. The first of these, the common rail system, has already been described. The second, the jerk pump system, will now be discussed.

The elements of one type of jerk pump system are shown in figure 3. A cam-operated *piston pump* is provided for each individual cylinder of the engine. This pump times and meters the fuel for each injection, it raises the fuel pressure above the compression pressure reached in the engine cylinder, and it supplies the energy required for atomization of the fuel. A surge of fuel is delivered to the *injection nozzle* at high pressure, and as the fuel emerges from the spray orifices into the engine cylinder, the sudden drop in pressure releases energy and atomizes the fuel. Associated equipment includes a *filter* (not shown) to insure against possible entry of dirt into the injector, and a *gear pump* to provide for positive delivery of fuel to the injection pump intake. The engine governor operates to adjust the effective displacement of the injection pump. Manual adjustments of speed are effected by adjustment of the governor linkage.

In figure 3 the injection pump and the injection nozzle are constructed in a single unit.

Types of injection pumps.—All of the various methods of constructing variable-delivery pumps which were discussed in connection with air-injection systems have been employed for solid-injection systems. The field is now largely dominated by three general types of



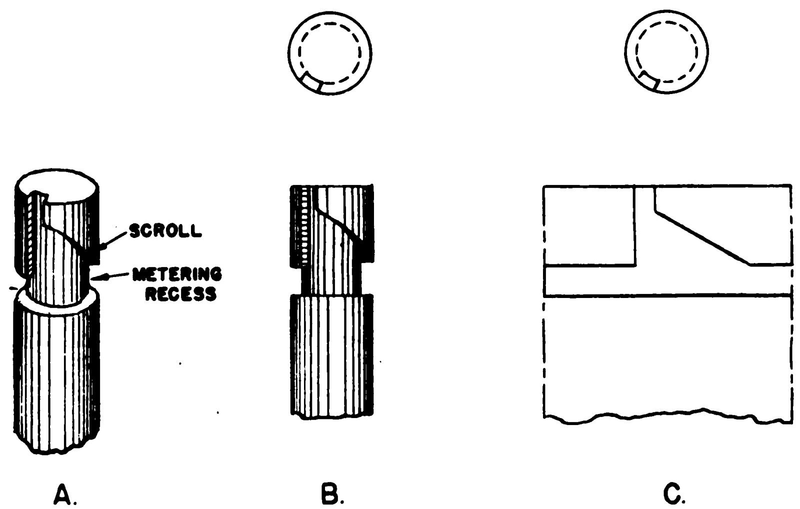
SOLID-INJECTION DIESEL ENGINE

Figure 3.

commercial pumps: the Bosch, General Motors (Winton), and Ex-cell-o. Pumps of these types are constructed by various manufacturers, and the individual designs differ somewhat from each other in non-essential details.

The Bosch injection pump.—The Bosch injection pump is a scroll bypass, cam-operated, constant-stroke pump. The piston is fitted very accurately in the pump cylinder with a clearance of less than 0.0001 inch, and a long enough leakage path of high resistance is provided so that packing is unnecessary. A scroll and a metering recess are milled in the plunger of the pump giving it the shape shown in the various views of figure 4. The metering recess is directly connected to the cavity in the pump cylinder above the plunger.

A Bosch type pump is shown in figure 5. The plunger is shown fully retracted and ready to commence the delivery stroke. The plunger is forced upward by the action of the pump cam and is returned to its initial position by the plunger spring. The upper limit of the plunger stroke is indicated by a dotted line. Filtered oil is supplied to the intake port under a pressure of 10 to 25 pounds per square

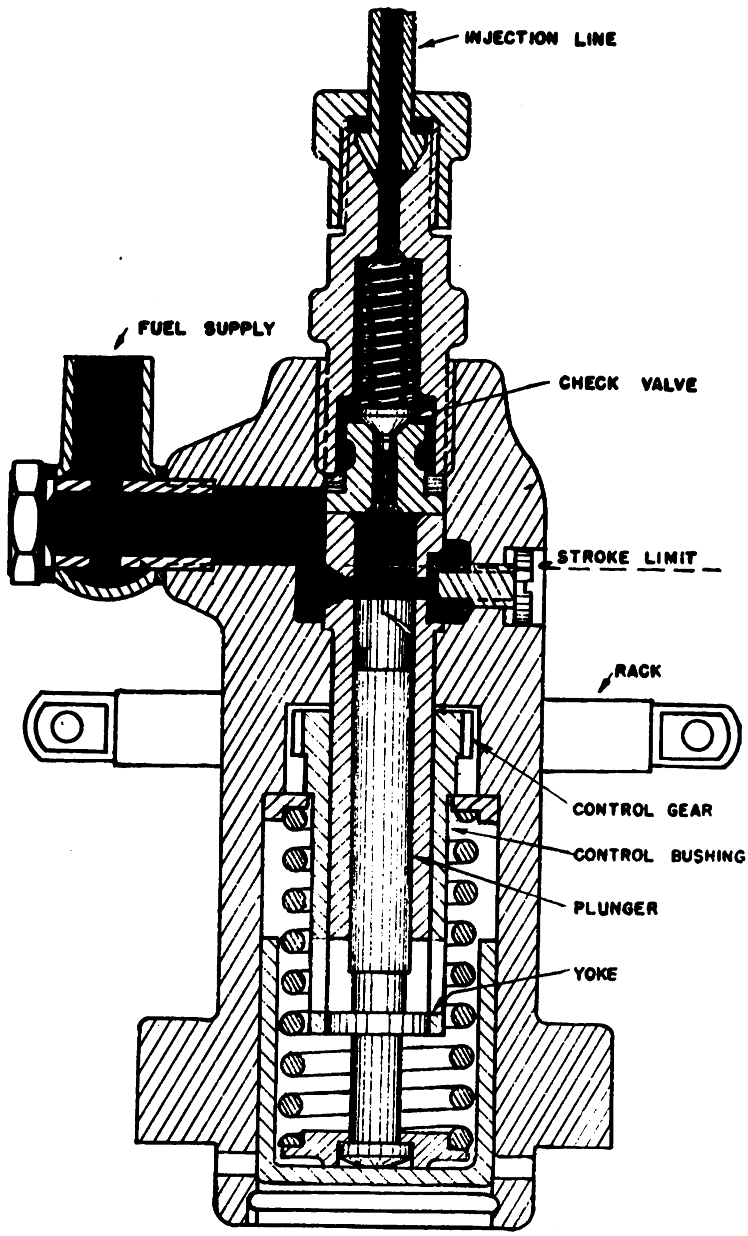


BOSCH INJECTION PUMP PLUNGER

Figure 4.

inch gage by the gear pump. This oil fills an annular cavity encircling the pump cylinder in way of the fuel ports. The oil is indicated by the solid black areas of figure 5.

The action of the pump plunger is detailed in figure 6. The pump is set for about 80 percent of maximum delivery. During the time that the piston is at the bottom of its stroke, oil enters the cylinder through the two ports and fills the space above the piston. As the plunger moves upward, oil is displaced from the space above the plunger and flows back out the ports as indicated in figure 6a. This action continues until the plunger covers both ports, as in figure 6b. The oil in the cylinder and in the fuel lines is compressed until enough pressure is developed to lift the delivery valve against its spring. The fuel is then delivered to the engine as indicated by the arrows. Delivery of fuel continues during the up stroke of the plunger until one of the ports is uncovered by the upper land of the metering recess. This terminates delivery and, as indicated in figure 6c, during the remainder of the up stroke of the plunger the fuel above the plunger flows through the metering recess and out the port.



BOSCH INJECTION PUMP

Figure 5.

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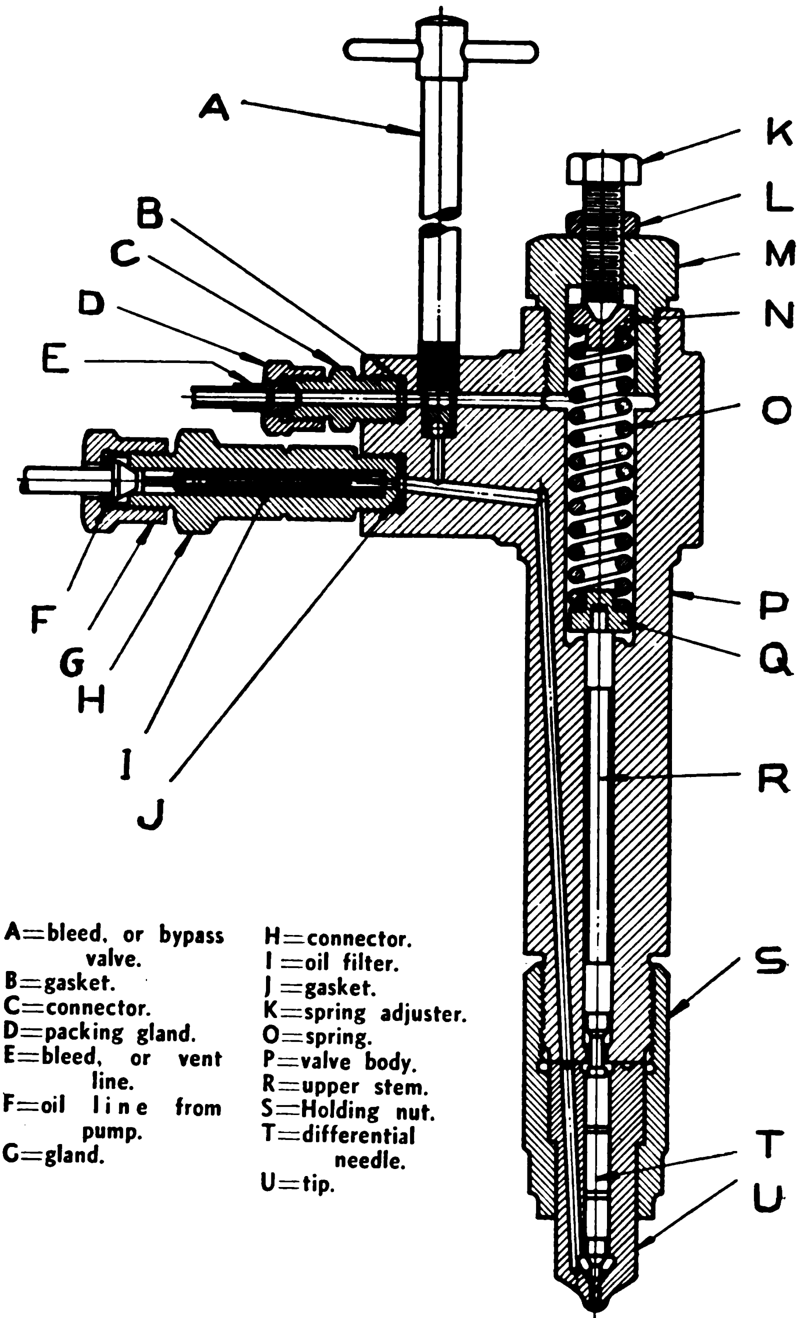
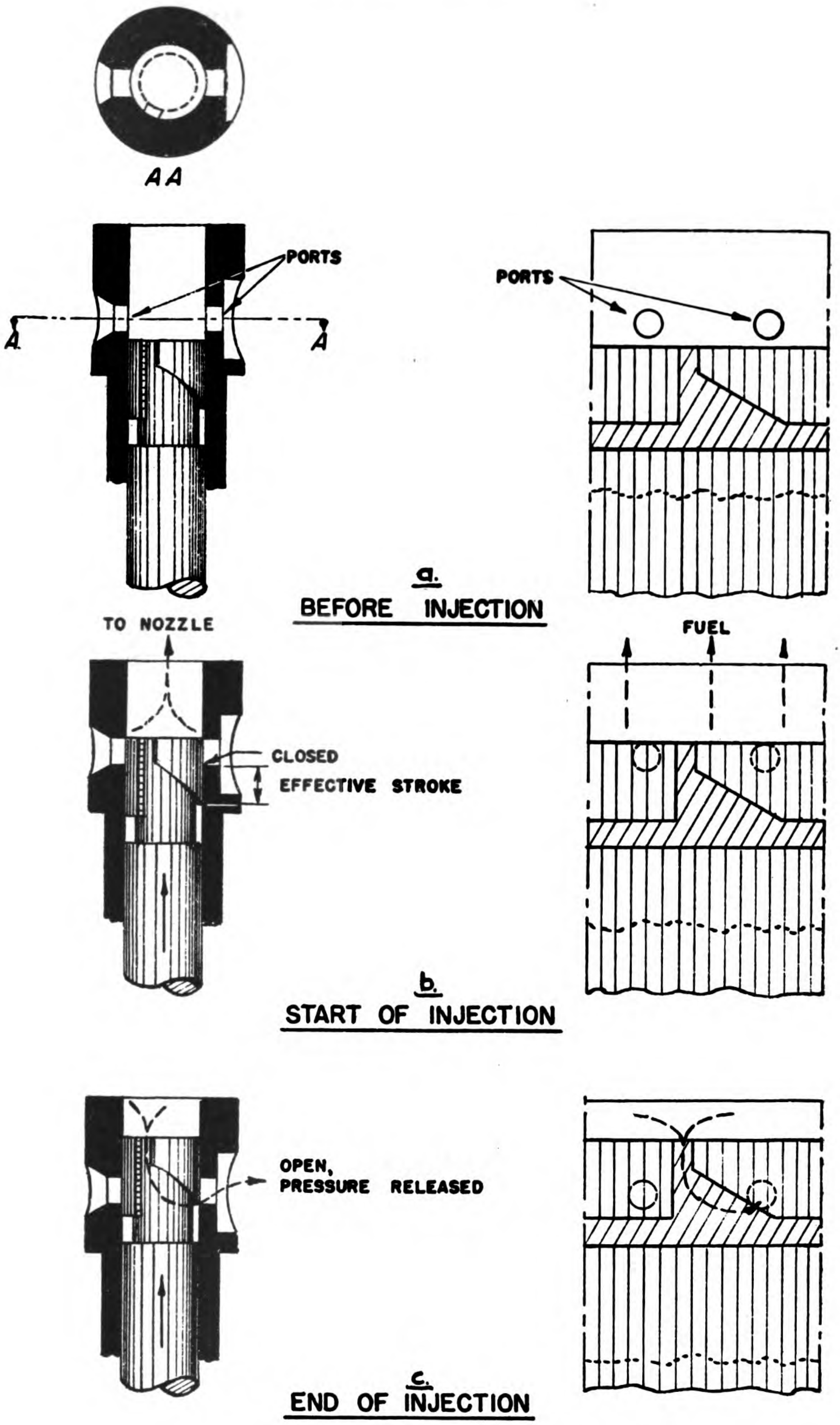


A=bleed, or bypass valve.
B=gasket.
C=connector.
D=packing gland.
E=bleed, or vent line.
F=oil line from pump.
G=gland.
H=connector.
I =oil filter.
J =gasket.
K=spring adjuster.
O=spring.
P=valve body.
R=upper stem.
S=Holding nut.
T=differential needle.
U=tip.

Bosch Spray Valve.



ACTION OF BOSCH INJECTION PUMP

Figure 6.

Two types of drawings have been used in figure 6 to illustrate the pump action. The column to the left is a series of random sections with one top view included at the head of the column. The column to the right is a series of developments of the piston showing the same positions of the plunger as the corresponding random sections.

The delivery of the pump is adjusted by rotating the pump plunger so that the scroll uncovers the bypass port later in the stroke for greater delivery, as in the left hand column of figure 7, or earlier in the stroke for smaller delivery. The plunger may be rotated to the position shown in the column at the right of figure 7, in which case the right hand port is never covered and no fuel is delivered to the engine.

Rotation of the pump plunger may be accomplished while the engine is operating. Longitudinal movement of the control rack acts through the control gear to rotate the control bushing. The lower end of the control bushing is forked, and the plunger yoke, which is rigidly attached to the plunger stem, rides in the control bushing fork as the plunger reciprocates. The engine governor and hand throttle act to position the control rack, and thus control the delivery of fuel to the engine.

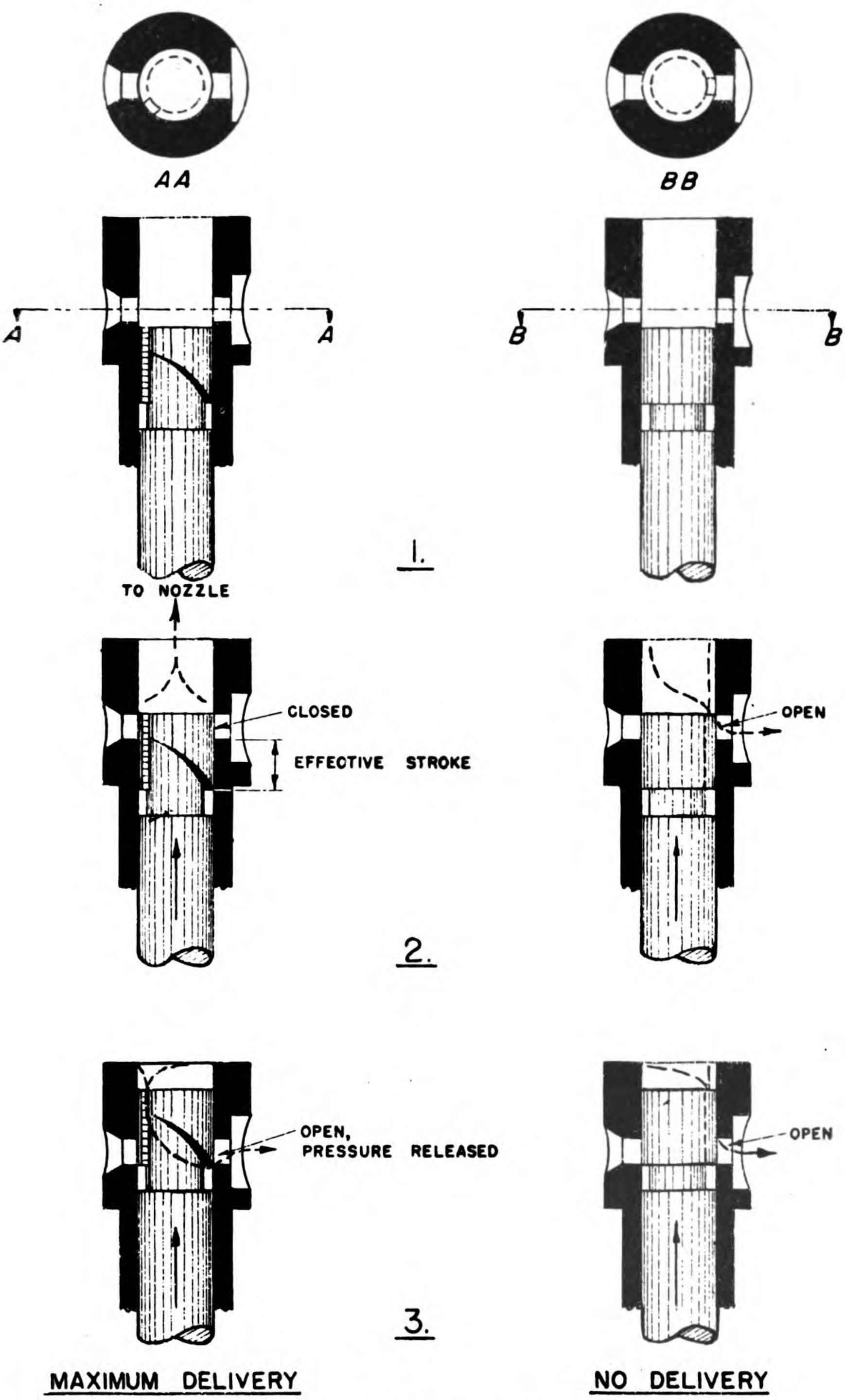
The Bosch pump begins its effective stroke at a constant point in the engine cycle and terminates it when the required amount of fuel has been delivered. This is desirable for a constant speed engine, but not ideal for a unit which operates at variable speed. Accordingly, the injection should be timed for best operation at the most common engine speed.

There is a slight leakage of fuel oil past the pump plunger, amounting to about 0.2 percent of the full effective pump displacement. This leakage provides for the lubrication of all of the parts of the pump assembly, and no other lubricant is required within the main housing.

An individual pump is supplied for each cylinder of the engine. These pumps are usually assembled within a single sealed unit mounted at the side of the engine. A fuel line leads to the injection nozzle of each individual engine cylinder.

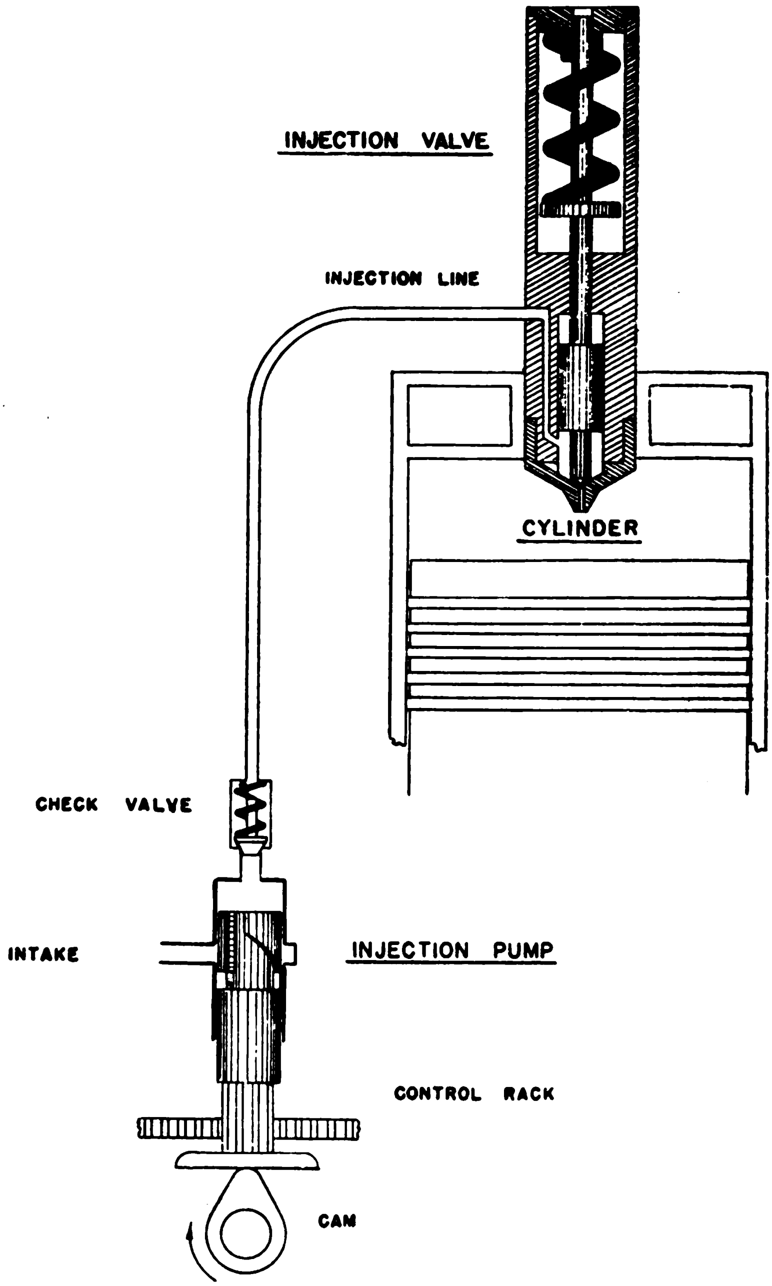
The General Motors (Winton) pump.—The General Motors pump is a modification of the earlier pumps constructed for Winton Diesel engines. The pump is an integral part of the General Motors unit injector, and is shown in place in the complete assembly in figure 8.

The General Motors pump is similar in principle to the Bosch pump, but differs greatly from it in detail. Oil is supplied under a pressure of 15 to 25 pounds per square inch gage by a gear pump and enters the injector assembly through the fuel supply connection at the right. As the fuel enters it is passed through a fine mesh screen which protects the pump from accidental entry of dirt with the fuel.



ACTION OF BOSCH INJECTION PUMP

Figure 7.



BOSCH INJECTION SYSTEM

The fuel is delivered through a passage in the injector body into an annular space which encircles the pump cylinder in way of the ports. This annular space connects with a second passage in the body of the injector which is in turn, in some installations, connected to a fuel return line. If the return line is fitted, there is a constant flow of oil past the pump which acts to prevent any possible accumulation of air in the unit and also provides for maintaining a definite fuel supply temperature.

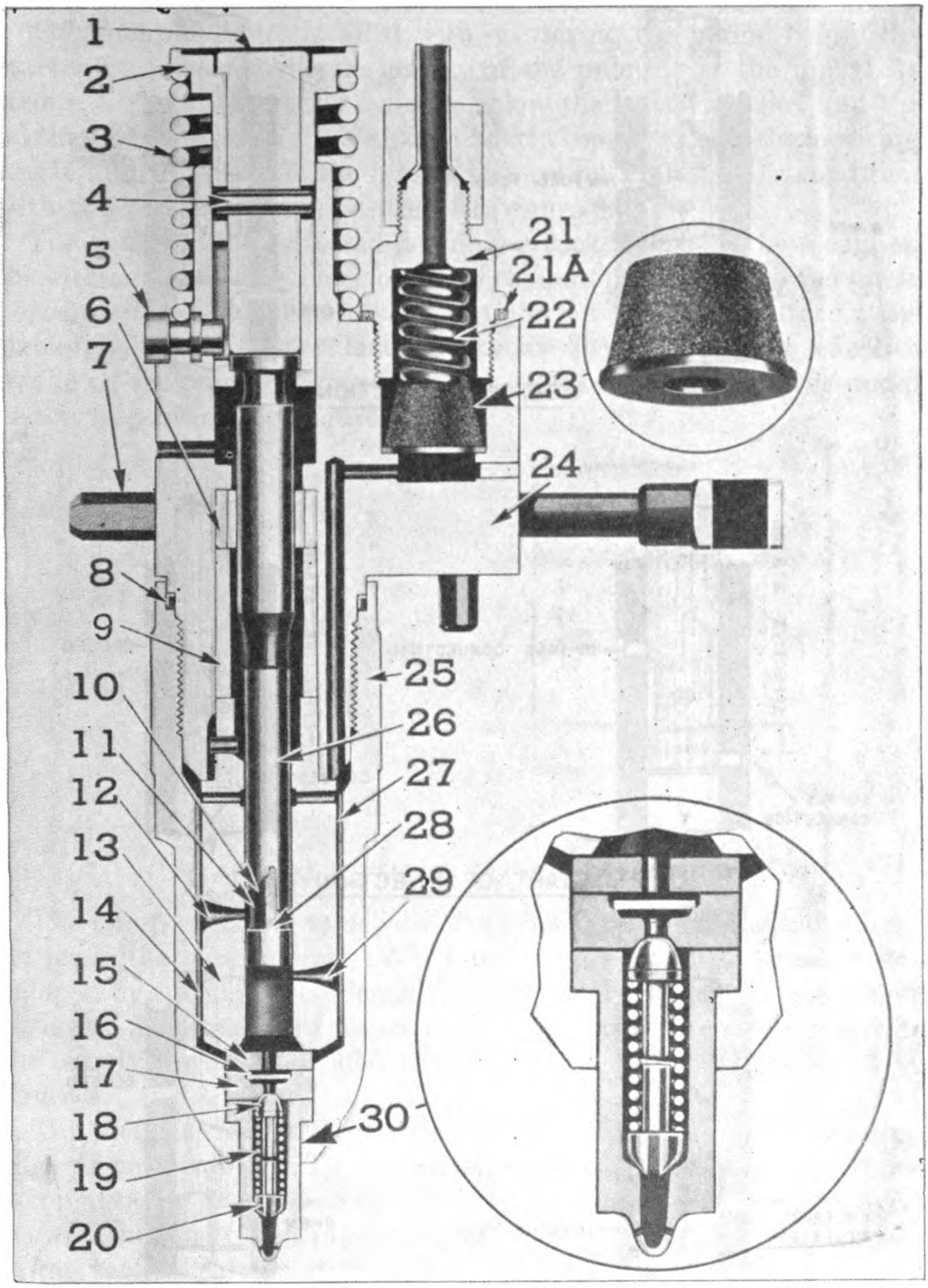
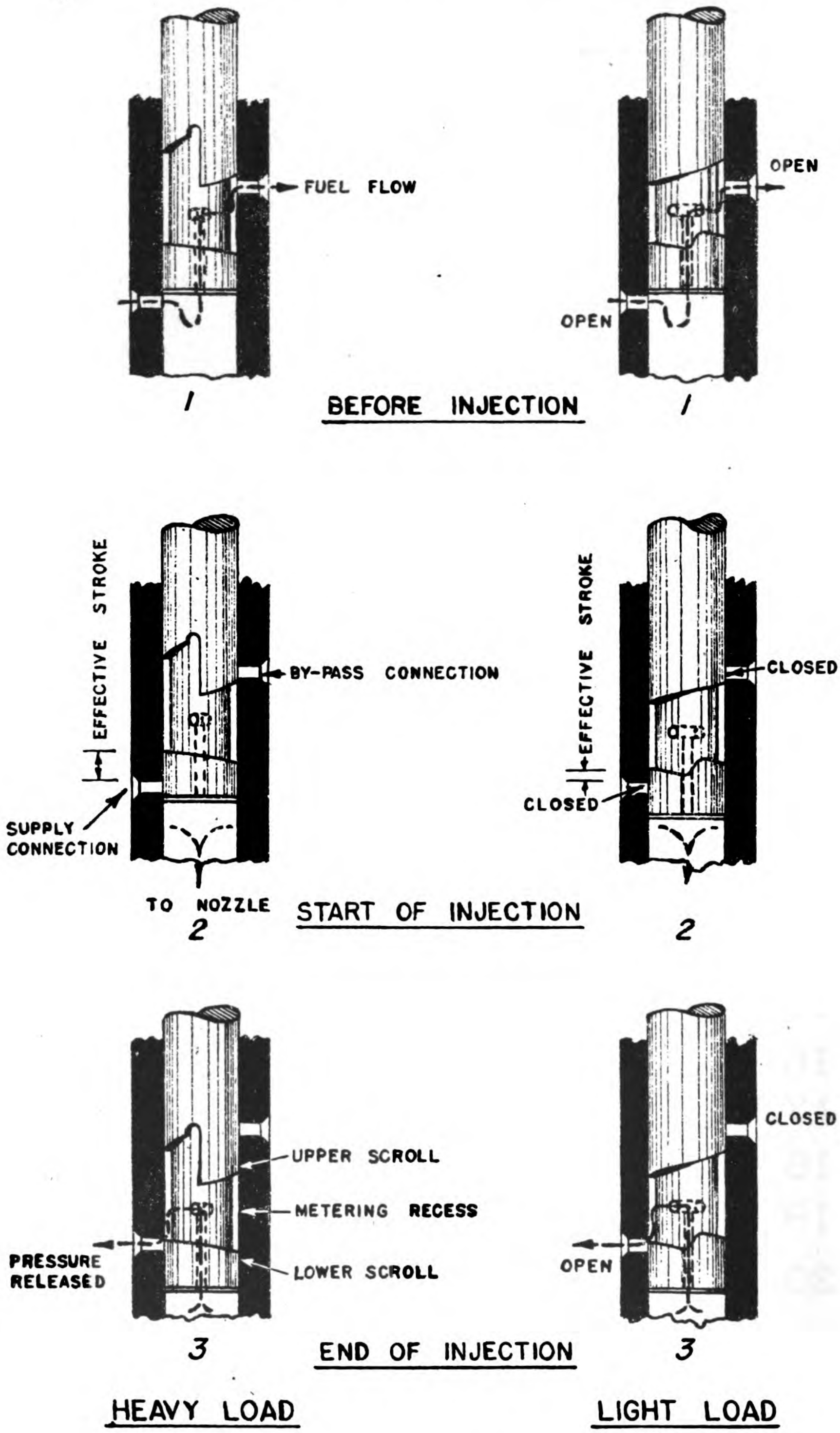


Fig. 8—General Monors Unit Injector

1. Follower
2. Follower Guide.
3. Plunger Follower Spring.
4. Follower Pin.
5. Follower Stop.
6. Gear.
7. Control Rack.
8. Seal Ring.
9. Spacer.
10. Upper Helix.
11. Metering Recess.
12. Upper Port.
13. Bushing.
14. Spill Deflector.
15. Flat Check Valve Seat.
16. Flat Check Valve.
17. Spherical Check Valve Seat.
18. Spherical Check Valve.
19. Check Valve Spring.
20. Check Valve Stop.
21. Filter Cap.

21A. Copper Gasket.

22. Filter Spring.
23. Filter Assembly.
24. Injector Body.
25. Injector Nut.
26. Injector Plunger.
27. Fuel Cavity.
28. Lower Cut-Off Helix.
29. Lower Port.
30. Spray Tip.



ACTION OF PUMP IN G-M UNIT INJECTOR

Figure 9.

The pump cylinder is filled with oil during the period before the start of injection while the piston of the pump is at the top of its stroke. The space in the cylinder below the piston is filled and the oil then flows through the T-shaped hole in the piston into the metering recess and out through the bypass connection. This is the condition with the pump piston in position 1 in figure 9.

The pump piston is forced down by a rocker actuated by a cam on the engine shaft. Delivery of oil to the pump ceases when the lower portion of the piston covers the supply port. A little later in the stroke, when the upper land covers the bypass port, the effective stroke of the pump commences. This is the condition with the pump piston in position 2 in figure 9.

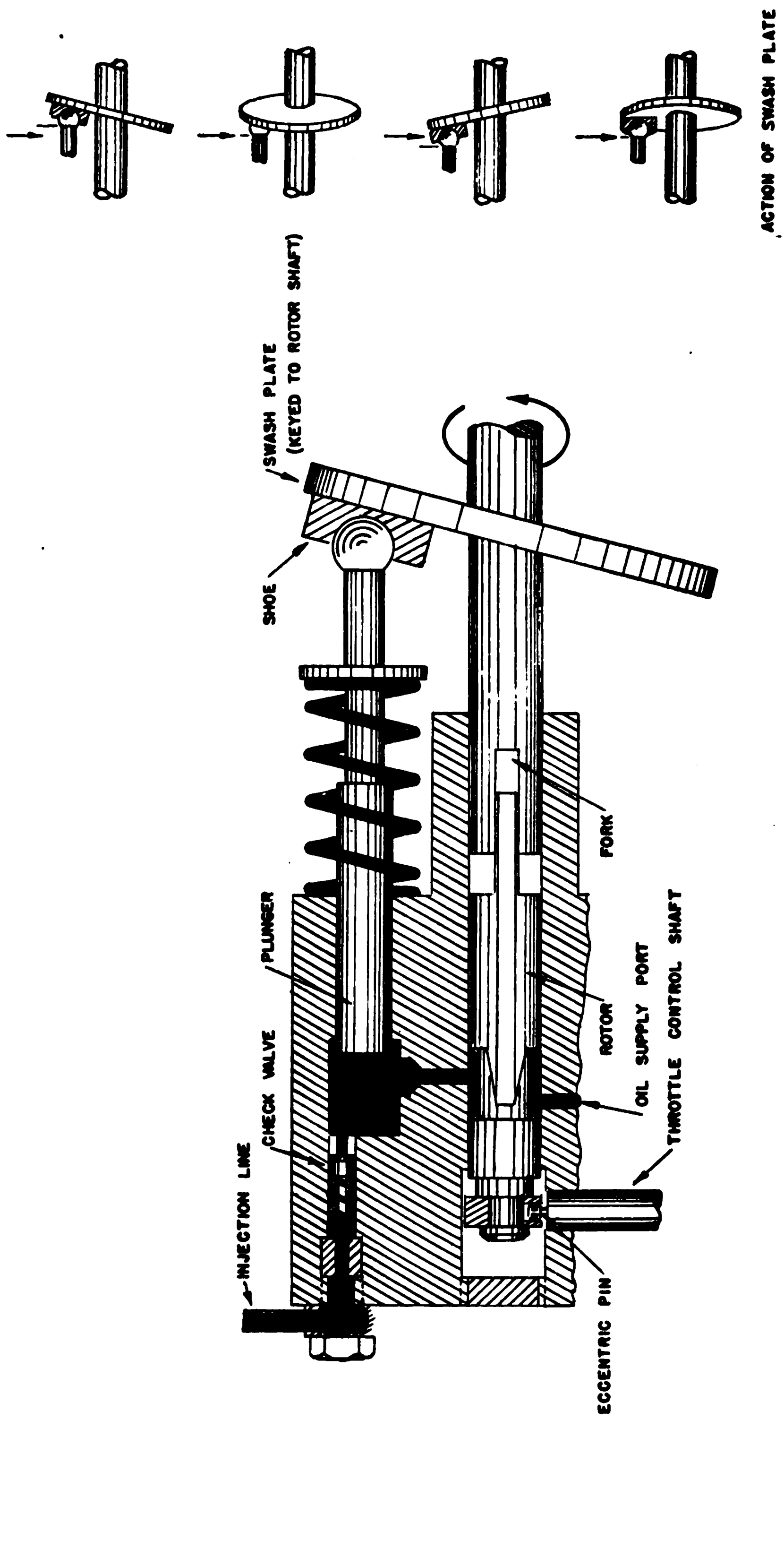
A. B.

G-M INJECTION PUMP PLUNGER

The pump continues to deliver oil to the nozzle until the lower land uncovers the supply port. When this occurs the high pressure developed by the pump is released, and for the remainder of the piston stroke oil flows back into the annual space around the cylinder through the supply port as indicated with the pump piston in position 3 in figure 9.

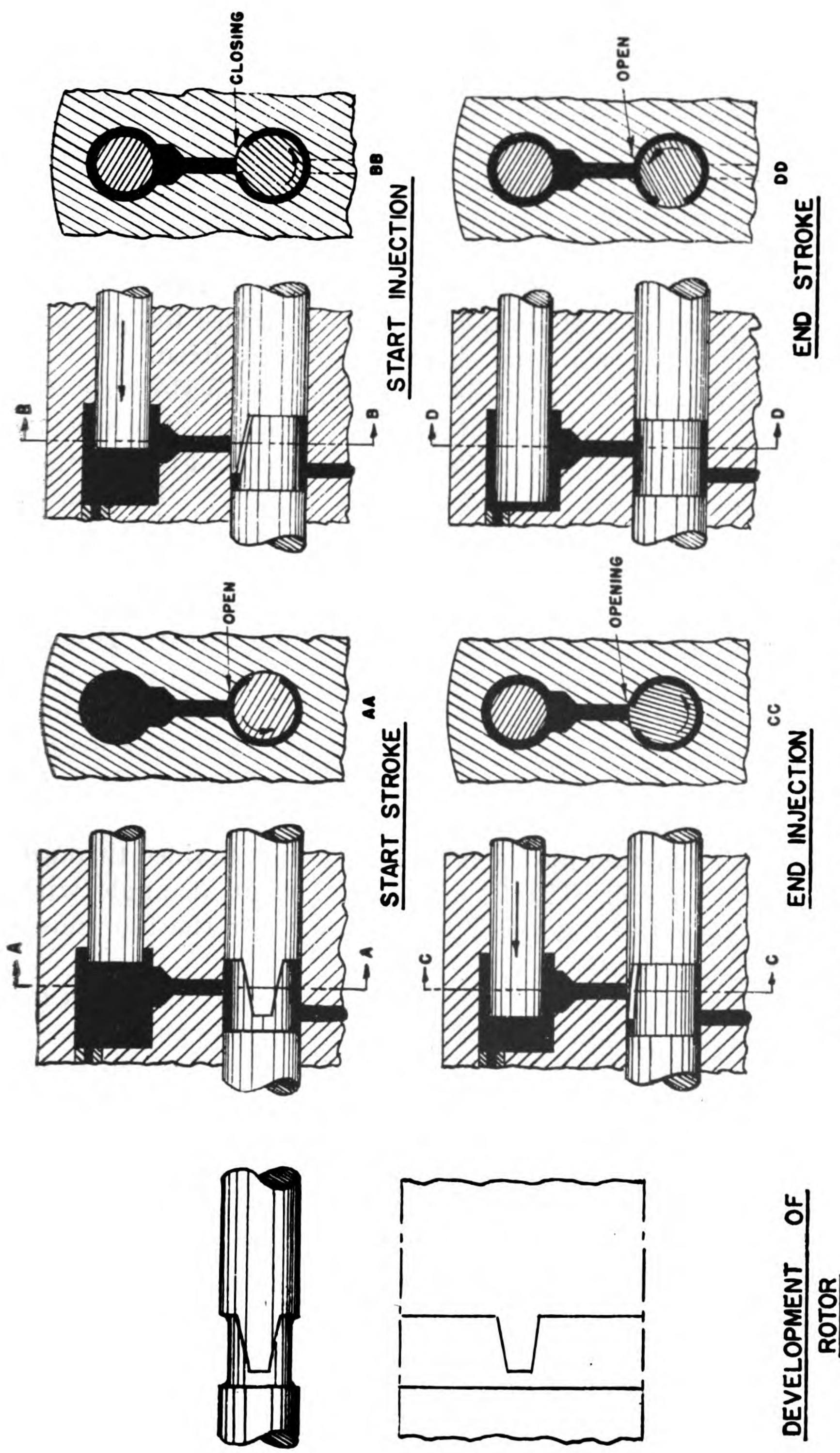
The metering recess is bounded at both the upper and the lower edge of helical land. To vary the time at which the effective stroke is commenced and the time at which it is terminated, provision is made for rotating the pump plunger in the cylinder. The three drawings in the vertical column at the left of figure 9 show the pump set for a heavy load on the engine. The other column of drawings shows the pump set for a light load on the engine.

The rack and the gear used to turn the plunger are shown in figure 8. The section of the pump plunger rod in way of the gear is shaped as an irregular heptagon. The plunger makes a neat sliding fit in a hole of the same shape in the gear, so that the plunger is free to move vertically inside the gear but is not free to rotate independently of the gear. The engine throttle and the governor are linked to the racks of the injectors installed in each of the engine cylinders and move the racks in unison.



SECTION OF EX-CELL-O PUMP (SCHEMATIC)

Figure 10.



ACTION OF EX-CELL-O PUMP

Figure 11.

The start of the effective pumping stroke is under the control of the upper land and the termination of the effective pumping stroke is under the control of the lower land bounding the metering recess. This makes it possible to fix not only the quantity of fuel injected per stroke, but also the timing of the injection period to meet the requirements of the engine. The piston in figures 8 and 9 is of the type used in variable speed engines such as those in ships. Because the engine is expected to make its greatest speed while demanding the greatest power, the shape of the two lands provides for advancing the injection period as the pump delivery is increased. This inherent control is a design feature of considerable value. Once the plunger is constructed the injection timing characteristics are fixed.

The Ex-cell-o pump.—Figure 10 is a random section showing the essential working parts of an Ex-cell-o injection pump. The section plane is passed through the axes of the rotor cylinder and the plunger cylinder, and passes in front of all of the working parts. The pumping element is a smooth cylindrical piston which fits oiltight in the pump cylinder without packing. The compression cavity of the cylinder is larger in diameter than the piston, so that although the piston passes the intake port in its stroke, it does not seal it. During a portion of the plunger stroke the supply port is sealed by a rotary valve (termed the rotor), and while this port is sealed the pump delivers oil under high pressure to the fuel line. A spring loaded valve is provided to maintain pressure in the fuel line during the suction stroke of the pump.

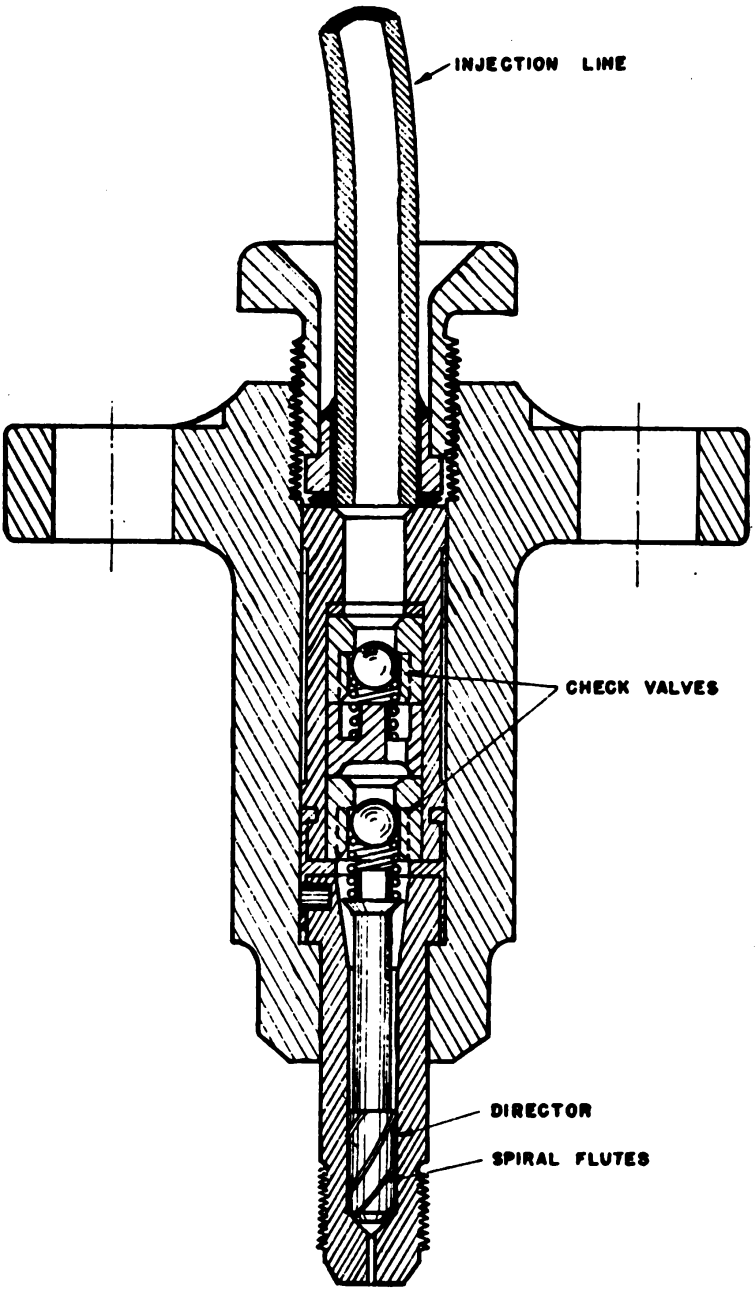
The reciprocating motion for the pump plunger is obtained through the use of a swash plate. This is a flat disk set at an angle with the shaft. The action of the swash plate is shown in the four small figures at the right of figure 10.

The views in figure 11 show side and end views of the working parts of the Ex-cell-o pump at four significant points in the cycle.

The land of the rotor is tapered, and the rotor is free to move longitudinally in the rotor cylinder to a position set by an eccentric pin in the end of the throttle control shaft. Moving the rotor to the right decreases the effective pump stroke, and moving the rotor to the left increases the effective pump stroke.

By a detail of construction not shown in the illustration, provision is made to turn the rotor with respect to the swash plate to adjust the injection timing. This adjustment may be made while the engine is operating.

Ex-cell-o injection pumps are usually made for multiple cylinder engines. All of the injection pumps for the engine are constructed in a single unit with the individual plungers (one for each engine cylinder) grouped around a single rotor. The engine governor, the



OPEN TYPE INJECTION NOZZLE

Figure 12.

gear pump, and a filter are included as integral parts of the assembly. Units of this type, with four or six plungers, are used on United States Navy motorboat engines. The pump is lubricated from the engine's forced-feed system.

INJECTION NOZZLES FOR JERK PUMP SYSTEMS

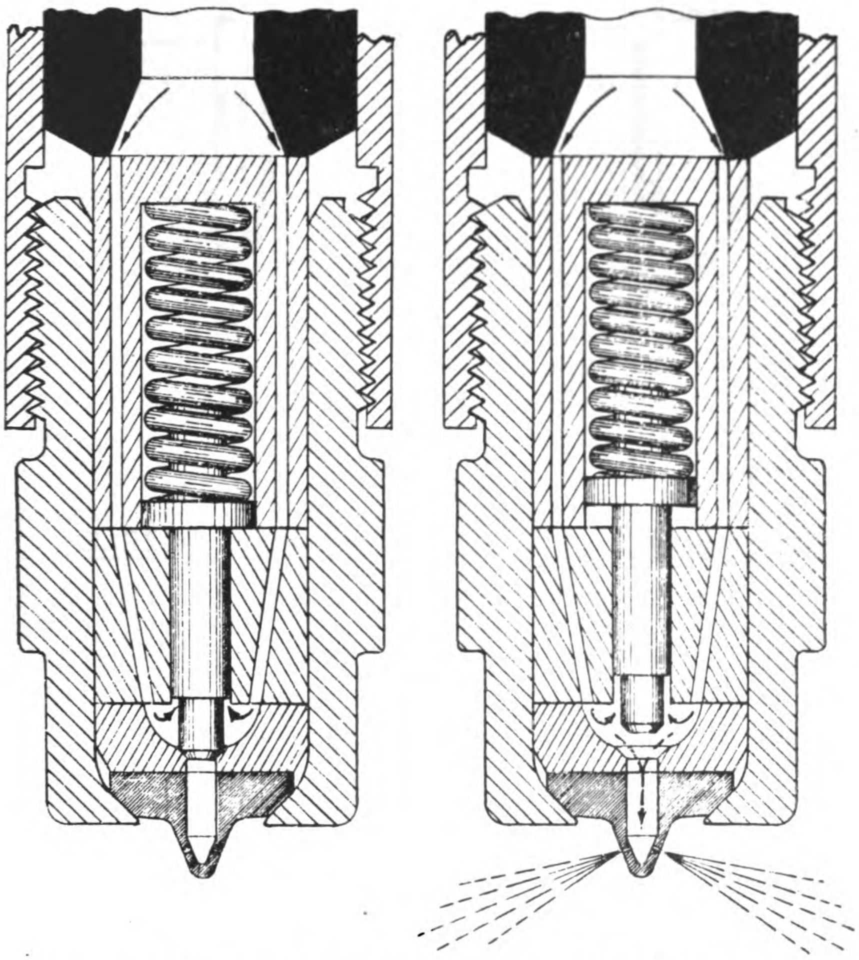
There are two principal types of injection nozzles for jerk pump systems; the open type, and the closed type.

Figure 12 shows one type of open injection nozzle. The surge of fuel delivered by the jerk pump opens the spring loaded check valves at a predetermined pressure, passes through the spiral flutes on the director in the nozzle assembly, and emerges with a whirling motion from the nozzle orifice into the engine cylinder. The spring-loaded check valves prevent dripping of fuel from the nozzle into the engine after the desired quantity of fuel has been injected. This design is used with low injection pressures (1,000 pounds per square inch gage) in connection with a high-turbulence combustion chamber construction. Due to the combustion chamber design, the air in the chamber is given a whirling motion, and this, rather than the nozzle action, is relied upon for atomization of the fuel. The air in the cylinder whirls in the direction opposed to the spray whirl.

Figure 13 shows the basic construction principles of a hydraulic injection valve of the closed type. Closed type nozzles are used on high-pressure ignition systems and now dominate the field. One of the principal advantages of the closed type nozzle is its unvarying spray characteristic. The valve is opened by fuel pressure and closed by a spring. When the valve is closed, the fluid pressure acts only on the shoulder of the plunger, but after the needle is lifted clear of its seat, the fluid pressure acts over the entire cross section of the plunger. Because of the differential action of the hydraulic pressure on the plunger, a greater fluid pressure is required to open the valve than is required to hold it open.

The Bosch type injection nozzles, used by many Diesel engine manufacturers, follow in principle the construction in figure 13. Four different types of tips which are used with Bosch type nozzles are shown in figure 14. These are (1) the Bosch single-hole type, (2) the Caterpillar single-hole type, (3) the pintle type, and (4) the multiple orifice type.

Single-hole nozzles are those in which the fuel jet is formed by a single circular orifice in the nozzle tip. The resultant jet is of conical shape, the angle of the cone varying between 4° and 15°, depending on the proportions of the parts of the nozzle. Nozzles of this type produce nonuniform sprays when the angle of the cone exceeds 12° if there are slight inaccuracies in manufacture. Spirally grooved



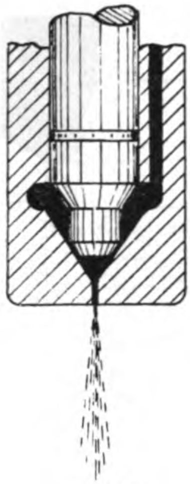
ACTION OF DIFFERENTIAL PRESSURE NOZZLE

Figure 13.

directors, similar to the one shown in figure 12, are used to obtain wider spray angles when necessary, but this construction is inferior to the pintle design. Single hole nozzles are used in precombustion chamber engines and in high-turbulence direct-ignition engines.

The Caterpillar design differs from the Bosch design in that the valve seat is flat in the former and conical in the latter. The advantage of the flat seat is that the spray disk may be removed and be replaced without the necessity of lapping the valve to fit the new disk.

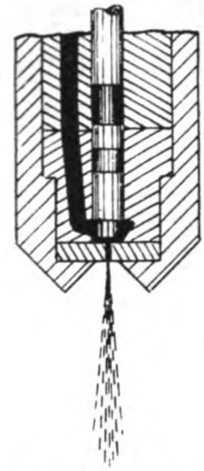
Pintle nozzles are characterized by a pin on the end of the valve which extends into the injection orifice to form an annular space. By suitably shaping this pin, it is possible to obtain a variety of spray formations. These vary from a hollow cylindrical jet of high pene-



BOSCH

ONE HOLE NOZZLE

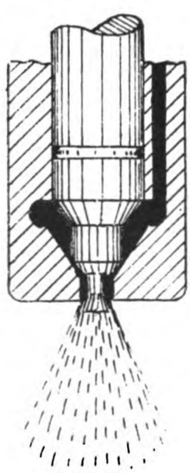
1.



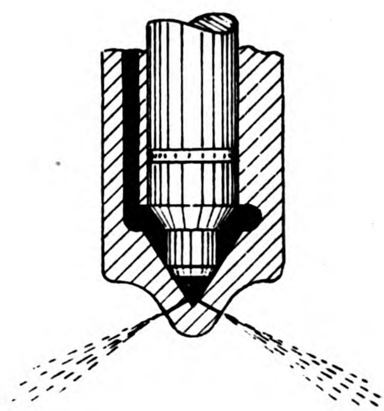
CATERPILLAR

ONE HOLE NOZZLE

2.



PINTLE NOZZLE



COOPER-BESSEMER

MULTIPLE HOLE NOZZLE

INJECTION NOZZLE TIPS

Figure 14.

trating power to a tapered spray with an angle varying from a few to as many as 60°. Pintle nozzles have very desirable characteristics and, if suitably designed, may be used to replace any other type. Their

disadvantage is that they are relatively difficult to manufacture and to maintain.

Multiple-hole nozzles, as their name implies, are similar to the single-hole nozzles except that two or more orifices are provided to distribute the fuel more uniformly in the combustion space. They are commonly used in plain combustion chamber engines, particularly those of large bore.

INJECTION PUMP AND VALVE INSTALLATION

In the case of both the Bosch and the Ex-cell-o injection systems, all of the injection pumps are mounted together in a single unit installed at a convenient position on the engine. The individual pump units are connected to their associated spray valves with high pressure fuel lines. These lines are of small bore, and are made of steel or noncorroding alloys. Fuel oil is appreciably compressible and is highly elastic. Elastic pressure waves are set up in the fuel lines during injection, and this results in the pressure at the nozzle end building up by sudden increments during the injection stroke of the pump and dying out by definite increments after the pump stroke is terminated. Fuel is injected into the engine at an irregular rate during the injection period, and after the delivery of fuel by the pump has ceased a series of spurts of fuel are ejected from the spray valve as the pressure waves travel from end to end of the fuel line. This irregularity of injection is detrimental to efficient engine operation. Check valves are installed at both ends of the fuel lines to minimize the effect, but these are not entirely effective.

In order to obtain equal injection in all cylinders and similar injection rates, it is necessary to make all fuel lines on an engine of equal length. The lines must be made as long as those required for the cylinder farthest from the injection pump assembly, and the excess tubing is stowed by forming loops in the lines to cylinders nearer the pump assembly.

A better solution to the problem is obtained by eliminating the fuel lines by constructing the pump and injection valve in a single assembly, the so-called unit injector construction. The General Motors unit injector, shown in figure 8, is the outstanding example of this type.

5. MAINTENANCE OF INJECTION SYSTEMS

The correct and proper functioning of a Diesel engine depends on systematic maintenance of the injection system. Three things are required: (1) the system must be kept clean, (2) the injection timing and the balance between cylinders must be accurate, and (3) nozzles and check valves must be maintained in correct adjustment. The first requirement demands primarily that only clean fuel reaches the injection system. In nearly all cases, the fuel oil is filtered before it is fed

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into the fuel supply line leading to the injectors. The filter types will be considered in more detail in the following chapter. Filters must be inspected periodically, and when an appreciable amount of dirt has been collected, the unit must be cleaned or provided with a fresh filter cartridge.

Adjustment of injection timing with air-injection engines is generally provided for by constructing the injection valve cam with an adjustable nose. The cam nose is loosened and moved slightly to advance or retard the time at which injection starts. The point of opening of the injection valve may be approximately determined by noting when the cam following roller becomes wedged so that it cannot readily be turned by hand. A better method is to put pressure on the injection line, remove the indicator cock from the cylinder head, and listen for the hissing of air entering the engine. Still another method is to install a dial micrometer (dial gage indicator) in such a position that the motion of the needle valve stem may be measured directly. This latter method is perhaps the most desirable of the three because it permits direct measuring of the time of opening, the amount of opening, and the time of closing of the valve.

Adjustment of timing with Bosch type pumps is accomplished by rotating the pump cam shaft with respect to the engine cam shaft. An adjustable coupling is sometimes provided to facilitate this operation. Fine adjustments of timing often are made by adjusting the length of the push rod which actuates the pump plunger, thus varying the instant the plunger starts to move. Timing can be checked easily and accurately, as follows: Remove from the pump the fuel intake line, the delivery line, and the check valve and spring. Insert one end of a short length of ¼-inch rubber hose into the space normally occupied by the check valve and spring. As the engine is turned over slowly in the injection region, blow through the hose; the air will blow through the metering recess in the plunger and out the intake connection except during the period when the ports are sealed (effective stroke). By inserting the tip of one's little finger in the intake connection, the points where the air flow cuts off and resumes can be easily noted; these points mark the limits of the effective stroke, the start of which is constant on a Bosch pump. This starting point usually is taken as the point where injection actually starts. These points can be accurately located on the flywheel, and their angular rotation to top dead center noted.

> It should be noted that this is not necessarily the exact point at which injection of fuel into the engine commences. Before fuel may be injected into the engine cylinder, it must first be compressed to injection pressure and the check valves at both ends of the line must open. This takes an appreciable period of time

which varies with engine speed, length of injection lines, and check valve settings. The problem is further complicated by the fact that elastic compression waves travel from end to end of the fuel lines, causing erratic injection rates. These factors constitute the major faults of injection systems using separate pumps and valves.

Individual pumps in an assembly of pumps for several cylinders are timed with respect to other pumps in the unit by altering the length of the tappet which actuates the pump plunger. These tappets are threaded and are provided with an adjustable cap at one end. The cap is secured with a jam nut.

Unit injectors require two adjustments for correct timing. The injector rocker arm must be adjusted so that the proper stroke of the injector plunger is obtained. This action is checked through the use of a special fixture which permits exact measurement of the distance from the top of the plunger follower (the cap over the plunger and spring in figure 8, Diesel Engines) to some reference point on the valve body. Following this preliminary adjustment, the position of the rack with respect to the racks of other injectors on the engine is adjusted by lengthening or shortening the rack control link. These two adjustments are interdependent, as rotation of the plunger in the unit injector changes both the timing and the amount of fuel injected per stroke, while adjustment of the rocker arm also changes the injection timing.

Ex-cell-o units are provided with an adjustment for changing the position of the rotor with respect to the swash plate. This changes the timing of injection for all cylinders simultaneously. No provision is made for variation in the timing between cylinders. Due to the construction of the unit, such adjustment is unnecessary.

Most makes of injection nozzles for use with Bosch and Ex-cell-o pumps are provided with a means of adjusting the setting of the spring of the hydraulically operated valve in the nozzle. This adjustment changes the pressure required to open the nozzle, and slightly affects the amount of fuel injected for a given effective stroke of the pump. A test stand with a hand-operated pump and a pressure gauge in the injection line is used. The preliminary compression of the nozzle spring is adjusted until the nozzle opens at the pressure recommended by the manufacturer.

The experience of the forces afloat indicates that solid injection pumps and nozzles should never be torn down for repair unless performance on engine or test stand is defective. When necessary, however, the cleaning job should be entrusted only to the best mechanic available, because careless or inexpert handling of the parts inevitably results in serious damage. Operating clearances for a General

Motors injector are about 0.00008 inch, and the machining tolerance is of the order of 1 percent of the thickness of the paper in this book.

The first requisite for the cleaning job is to fix a clean working space. The bench should be covered with smooth-finish wrapping paper. A can of clean Diesel fuel or kerosene should be provided for washing the parts, and a disc of hard paper should be placed in the bottom of the can to protect the nozzle parts from scratching. One nozzle at a time is disassembled, washed free of carbon, and inspected. Carbon may be scraped from the outside of the nozzle, but care must be exercised not to mar the edges of the orifices in the burner plate or the ends of the pins of pintle nozzles. Carbon tetrachloride or acetone can also be used for cleaning pump and nozzle parts.

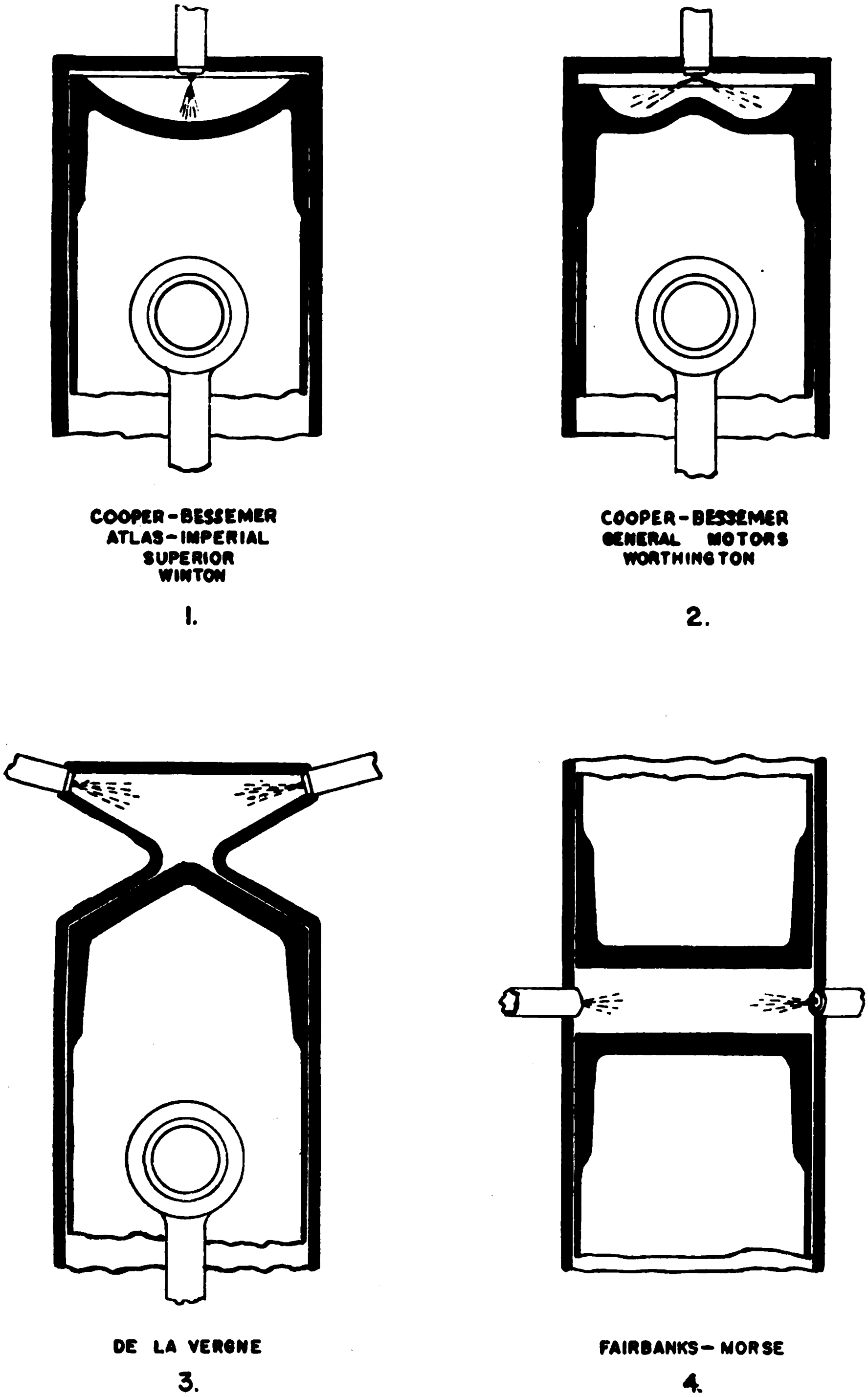
Reaming tools and special drills are provided for cleaning the holes of nozzles. No drills except those provided by the manufacturer should be used. The drills are operated by hand, the small ones used for cleaning the spray orifices being held with a small chuck called a "pin vise." It is essential that when this operation is performed, that only the foreign matter is removed. Particular care must be exercised to avoid burring the metal.

Valve seats for *air injection* valves may be reground if required. The finest grade of alumina powder is suitable for the purpose. Final surfaces are obtained by grinding the valves and seats with Diesel fuel as lubricant and *with no compound.* Valve seats for solid injection nozzles cannot be ground without special equipment, and should be replaced when worn.

The parts of solid injection pumps and nozzles must never be put together dry. They must first be lubricated with clean Diesel fuel oil.

After reassembly, nozzles and injection pumps should be tested before the engine is run. A special "popping fixture," which consists of a frame and an operating lever, is provided for testing unit injectors. The pump is filled with fuel oil from an oil can, and the injector is operated by hand with the aid of the operating lever. A test stand is sometimes provided for testing nozzles for Bosch and Ex-cell-o type injectors. If no test stand is available, the nozzles may be tested using the pump in place on the engine. In this case, the engine must be cranked at approximately starting speed.

It is extremely important when testing injection nozzles that the spray tip is so enclosed that the operator is protected. The jet velocities are very high, and oil will readily penetrate the skin, causing "burns" which usually lead to blood poisoning. The tip should be covered with a glass beaker or a similar transparent cover.



OPEN COMBUSTION CHAMBERS

Figure 1.

COMBUSTION SYSTEMS

Open combustion chamber engines.—In all air-injection engines and in most large solid-injection engines the fuel is injected directly into a space in the top of the engine cylinder. This system is practical for engines of slow and moderate speed which are equipped with single-hole or with pintle nozzles, and is also satisfactory for high-speed engines equipped with multiple orifice nozzles operated at high pressure.

Because of the high-compression ratios of Diesel engines, the volume of the combustion space is small. It is generally necessary to shape the combustion space to provide enough distance for injection without the spray impinging on the piston or the cylinder walls. This may be accomplished by shaping a cavity in the piston head, or by providing a dome in the cylinder head. Several examples are shown in figure 1.

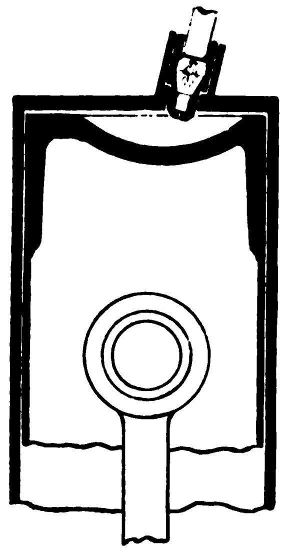
Intake air ports are frequently so arranged that the air is given an initial swirling motion as it enters. This persists during the compression stroke and assists in mixing the fuel and air.

The majority of engines use one injection nozzle per cylinder. Examples 3 and 4 of figure 1 show types which use two injection nozzles. In example 4, the spray is injected at an angle with the radial line passing through the injector nozzle. This gives the spray a whirling motion in the cylinder.

Open combustion chamber engines develop high thermal efficiency, start with relative ease, develop a high mean effective pressure, and have a simple cylinder head design. The major disadvantages are that a high fuel injection pressure is required (5,000 to 20,000 pounds per square inch), a high-compression pressure is necessary (16:1), and relatively high-grade Diesel fuel is necessary.

Precombustion chamber engines.—Small engines present a difficult combustion problem. Because of the relatively large ratio of surface area to cylinder volume, there is a tendency for the engine to run cooler than a large engine. This makes it necessary to produce a very well atomized injection spray. This may be accomplished by using a high injection pressure, but unless the spray orifices are very small, the spray will have a high penetration and will impinge on the cool surfaces of the piston and the cylinder walls resulting in incomplete combustion and generally unsatisfactory operation. Another approach which reduces the injector design problem involves the use of a construction which develops sufficient turbulence to permit the use of a relatively coarse spray. The precombustion principle provides a solution of this type.

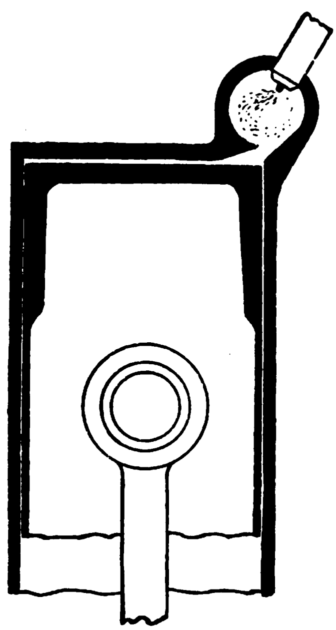
A precombustion chamber system is shown in figure 2. During the compression stroke, air is forced into the precombustion chamber. A



CATERPILLAR

PRE-COMBUSTION CHAMBER

Figure 2.



HERCULES

TURBULENCE CHAMBER

Figure 3.

high degree of turbulence is developed in the chamber. At the correct time in the cycle fuel is injected into the chamber and combustion commences. Combustion is only partially completed in the precombustion chamber, due to limited amount of air available in the chamber. As heat is liberated, the pressure in the chamber rises above the pressure in the cylinder and the hot oil vapor is ejected at high velocity into the main combustion chamber. The crater in the top of the cylinder is so shaped that a vortex forms which gives a thorough mixing of fuel and air in the main combustion space, and the fuel is burned to completion.

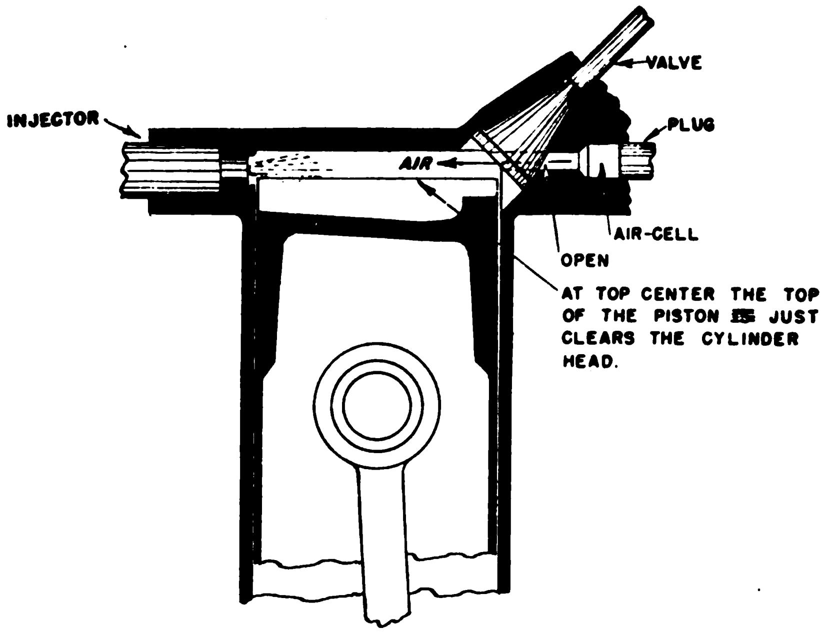
Heat transfer between the walls of the precombustion chamber and the cylinder head is poor enough to cause the chamber to run hot, and this aids in ignition of the fuel charge.

The precombustion chamber design permits low-fuel-injection pressure (<2,000 pounds per square inch), and makes possible the utilization of fairly viscous fuels. The fuel is burned completely. The disadvantages of the system are increased heat losses due to the greater surface-volume ratio than in a direct-injection engine, decreased mean effective pressure due to turbulence losses, and increased starting difficulty.

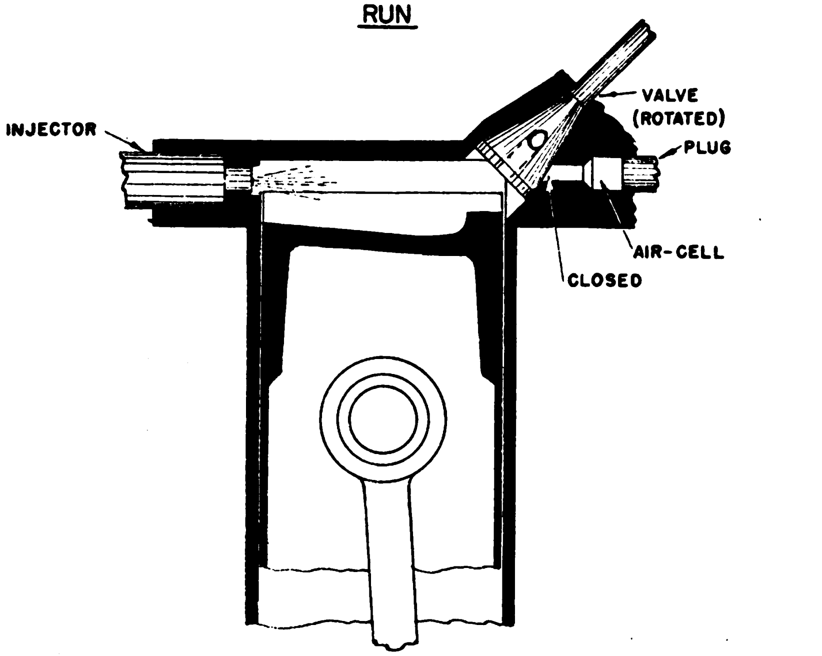
Turbulence chamber engines.—Another solution to the small engine combustion chamber design problem is provided by the turbulence chamber construction. This is similar to the precombustion chamber design, as may be seen by comparing figure 2 with figure 3. The volume of the turbulence chamber is great enough to accommodate nearly all of the air in the engine cylinder. The chamber is spherical in shape, and the entrance to the chamber from the engine cylinder is tangential. This causes a vortex to develop in the chamber during the compression stroke, and leads to rapid and complete mixing of the fuel and the combustion air. Combustion is practically completed in the turbulence chamber. As the engine cylinder moves on the expansion stroke, the hot combustion products are ejected from the turbulence chamber.

The advantages and disadvantages of the turbulence chamber construction and of the precombustion chamber construction are similar. Losses due to burning the fuel in a chamber partially isolated from the working cylinder are somewhat higher with turbulence chamber engines than with precombustion chamber engines. As the result, the mean effective pressures developed are lower.

Air-cell engines.—The third common solution to the small engine design problem is the air-cell system, illustrated in figure 4. The engine illustrated in this case is the Buda motorboat engine used by the Navy. The combustion space is provided by shaping the piston head as shown in the illustration. An air cell, or after chamber, is located



1.
RUN



2.
START

AIR-CELL OR AFTER CHAMBER

Figure 4.

opposite the injector nozzle. This chamber holds only a fraction of the total combustion air. Injection is commenced while the piston is at top center. As the piston moves downward, the pressure in the combustion space drops below that in the air cell, and the contents of the air cell are expelled into the combustion space at high velocity. This provides for adequate turbulence in the combustion space, and thoroughly mixes the air and burning fuel particles and vapor.

To assist in starting the engine, the after chamber is fitted with a rotary valve which may be turned to seal the after chamber from the main combustion space. This develops an extra high-compression pressure and temperature to offset the low engine temperature and reduces the difficulty of starting.

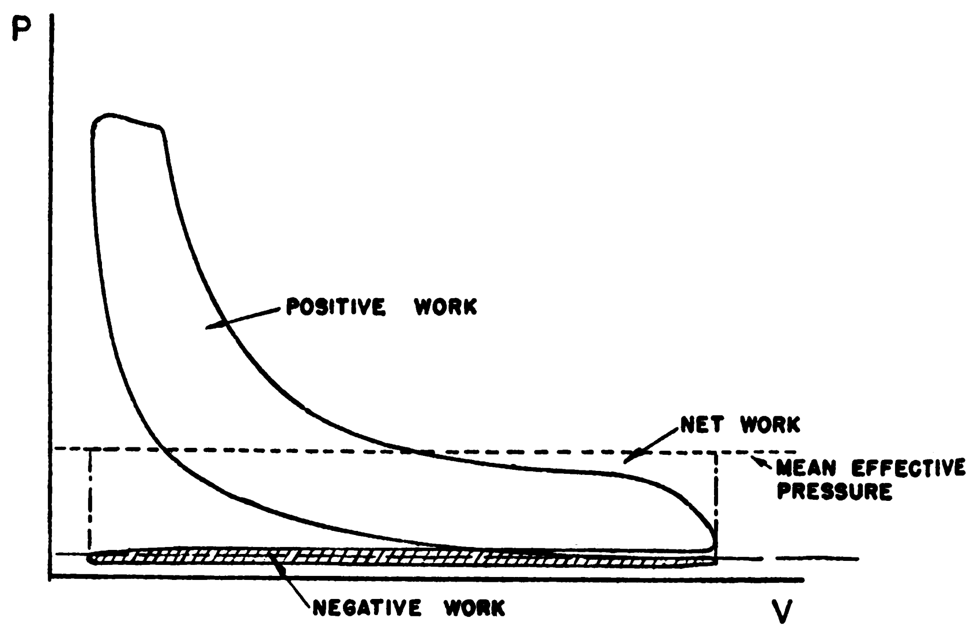
Due to the fact that combustion takes place in the working cylinder, air-cell engines can develop higher mean effective pressure than precombustion chamber or turbulence chamber engines. The construction is also better adapted to provide extra compression for starting.

ENGINE SPEED AND POWER

The power output of an engine may be calculated from the expression

$$\text{Horsepower} = \frac{PLAN}{33{,}000}$$

where P is the mean effective pressure, L is the length of the piston



INDICATOR DIAGRAM

Figure 5.

stroke in feet, A is the area of the cylinder, and N is the number of power strokes per minute. P and A must be in consistent units; that is, if P is in pounds per square inch, then A must be in square inches, or if P is in pounds per square foot, then A must be in square feet.

As explained in the chapter "Thermodynamics, 3," the work output of a cycle is proportional to the net area of a pressure-volume plot of the cycle. Such a plot may be determined experimentally through the use of an engine indicator. A typical indicator card for a four-cycle engine is shown in figure 5. The area enclosed by the pressure-volume trace during the compression and expansion strokes is considered positive and represents the power developed during these two strokes of the piston. The area enclosed by the pressure-volume trace during the exhaust and intake strokes is considered negative and represents the power absorbed from the flywheel (or from working cylinders) during these two strokes of the piston. The net area, the positive area less the negative area, represents the net power developed within the cylinder during the complete cycle. This is the indicated power developed by the engine. This area may be reduced to an equivalent rectangle with a base length equal to the piston displacement (bore times stroke), and the height of this rectangle is termed the *indicated mean effective pressure.*

Because of friction developed in the engine, the net power output of the engine is less than that calculated from the indicated mean effective pressure. The ratio of the actual horsepower output of the engine to the indicated horsepower is the *mechanical efficiency* of the engine. Using the known power output of the engine, the actual number of power strokes per minute, and the known values of bore and stroke, a value of mean effective pressure known as the *brake mean effective pressure* may be calculated.

An increase in engine power output may be obtained by an increase in P, L, A, or N. An increase in L or A results in a proportional increase in engine weight. An increase in P may require some increase in engine weight to increase the strength of certain engine parts, but power output may be increased more rapidly than engine weight increases. An increase in N may be accomplished with very little increase in engine weight. There has, therefore, been a big tendency toward improving engine performance in recent years by raising the mean effective pressure, and there has been an even greater trend toward higher engine speeds.

Two factors enter into considerations of speed classification of Diesel engines. These factors are crankshaft speed, and lineal piston speed. Engines running at 250 r. p. m. or less are considered low speed, those running up to 600 r. p. m. are considered medium speed, and those running at 600 r. p. m. or more are considered high-speed

engines. Maleev and Magdeburger (Trans. A. S. M. E., 1932) propose a characteristic (r. p. m.×piston speed feet per minute) ÷100,000. Low-speed engines are considered by them to be those with a characteristic less than 3, medium speed 3–9, high speed 9–27, and superhigh speed over 27.

For four-cycle engines, N=r. p. m.÷2, and for two-cycle engines N=r. p. m. for *each* cylinder, provided engines are *single acting*. Two-cycle engines therefore develop more power for a given bore, stroke, and speed than four-cycle engines. Because more power is demanded by a scavenging blower than is required for the two extra strokes of the four-stroke cycle, and because the four-stroke cycle clears exhaust gases from the cylinder more completely and develops a higher thermal efficiency, four-cycle engines are inherently more efficient than two-cycle engines.

6. DIESEL AUXILIARIES

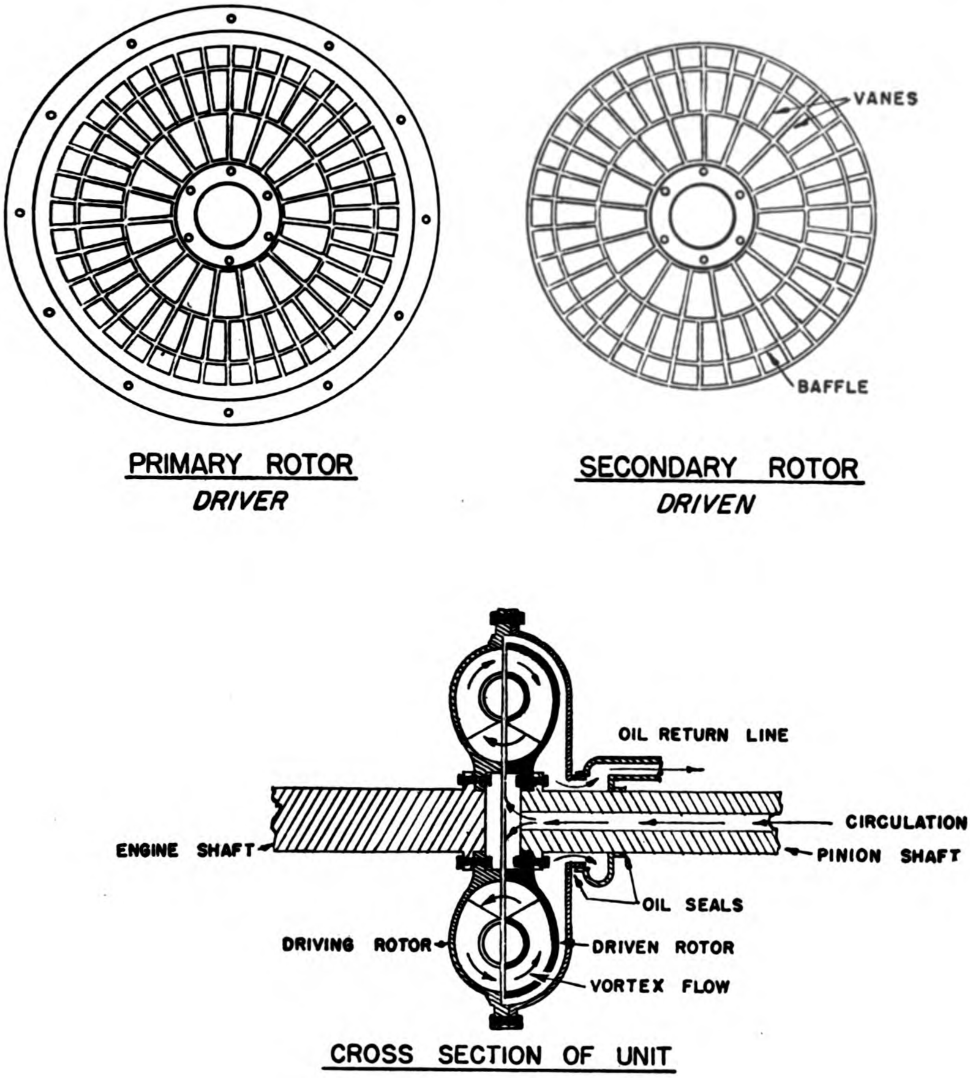
HYDRAULIC COUPLINGS

A hydraulic coupling, or fluid-drive unit, is detailed in figure 1. The two rotors are shaped so that together they form an annular ring of circular section. Both the primary and second rotors are partitioned with straight radial vanes and with baffles similar to the outer shell. Together the several parts of each rotor form a number of half rings as indicated in the cross section. The two rotors are provided with an unequal number of vanes to prevent generating pulsations in the speed of the driven rotor.

The unit is filled with oil through a hole in the secondary rotor shaft. In Navy units, the oil used in the coupling is the same as that used in the reduction gear assembly. A flow of oil is maintained through the unit for cooling it.

The oil in each of the rotors is acted on by centrifugal force and tends to flow outward from the center of the unit. The primary rotor runs faster than the secondary rotor by from 1 to 3 percent. The centrifugal force developed in the primary rotor therefore exceeds that developed in the secondary rotor, and the oil flows in loops through the ring-shaped spaces formed by the partitions. This flow carries the oil from the hub to the outer boundary of the primary rotor, across the small gap between the rotors, and returns it from the outer boundary to the hub of the secondary rotor whence it is directed across the gap to the starting point. This vortex flow is indicated by the arrows in figure 1.

The mass of oil in each of the rotors is turning about the main axis of the unit and possesses kinetic energy of rotation. Because the primary rotor turns faster than the secondary rotor, a unit quantity of oil in the primary rotor possesses more energy than a similar



HYDRAULIC COUPLING

Figure I.

unit quantity in the corresponding position in the secondary rotor. As oil flows from the primary rotor it carries the kinetic energy imparted to it by that rotor, and on being slowed as it enters the secondary rotor the excess of energy is delivered to the shaft attached to the driven rotor. The kinetic energy of rotation of a unit quantity of the oil is dependent upon the square of the distance from the axis of rotation to the center of gravity of the unit. Therefore, as the oil flows from the outer boundary to the hub in the secondary rotor, energy is continually released and imparted to the rotor shaft.

The torque output of a hydraulic coupling is identical to the torque input. The efficiency of the unit is equal to the ratio of the secondary rotor speed to the primary rotor speed.

Hydraulic couplings perform several important functions in a Diesel-powered ship. They prevent the transmission of torsional vibrations, thus separating the vibrating system from gears, shafting, propellers, etc., and making it possible to design the engine without reference to the specific installation. It protects the engine and gears from damage due to shock incident to fouling the propeller or to casualty to some part of the installation. It serves as a clutch to disconnect one engine of a multiple engine installation. (For this purpose special units are designed with provisions for rapidly draining the oil from the rotors.) It provides for rapid reversing (as previously described). It serves as a flexible coupling tolerant of a considerable degree of misalignment.

ELECTRIC COUPLINGS

Electric or magnetic couplings are in some cases substituted for hydraulic couplings to isolate the engine from the reduction gears, propeller, etc.

REDUCTION GEARS

As shown in the preceding chapter, the power which may be obtained from an engine of a given size is largely dependent on the engine speed. Propellers operate best at low speeds, and with direct drive installations propeller speed becomes the limiting factor. The design balance is such that the highest efficiency, and the least weight and cost are obtained by using engines of moderate speed driving the propellers through reduction gears.

Reduction gear ratios range from approximately 1.5:1 to 6:1. Many modern units use a gear ratio of about 2.2:1, operating the engines between 400 and 1,000 r. p. m. for engine powers up to about 2,000 horsepower. Some of the Maritime Commission C–2 cargo ships use engines running about 240 r. p. m. and driving the propellers about 90 r. p. m. through a 2.6:1 reduction gear.

Single reduction units, consisting essentially of a bull gear driven by a pinion, are normally used. Either single helical or double helical gears may be used. Single helical gears develop high end thrust, but are cheaper than double helical gears. The choice is determined by the least total cost of the unit, including thrust bearings.

ELECTRIC DRIVE

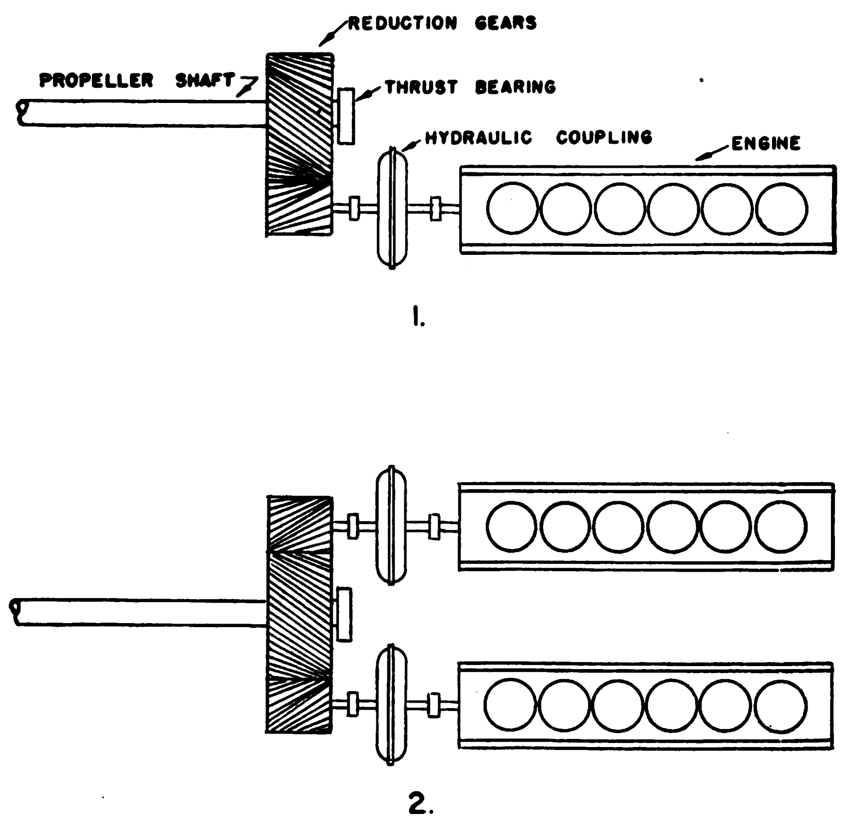
Diesel engines do not run well at less than one-third of their full speed. For applications which require high power at low speed,

some type of variable speed reduction unit is required. Diesel electric drive offers a satisfactory solution to the problem. Direct connected generators are driven by Diesel engines and furnish power for motors driving the propeller shafts. The best over-all design may be obtained by installing a reduction gear between the motor and the propeller shaft in order to reduce the size of the main drive motors. Part of the speed control is obtained by changing the engine speed, and part by electrical adjustments. Reversing is accomplished electrically.

Electric drives are used for large tug boats, net tenders, etc.

DIESEL ENGINE INSTALLATIONS

Single engines may be connected directly to the propeller shaft providing the engine operates at low speed. Higher speed engines must be connected through reduction gears or by an electric drive.



DIESEL ENGINE INSTALLATIONS

Figure 2.

Most multiple engine installations employ reduction gears. If reduction gears are employed, then nonrigid couplings are used. Figure 2 shows layouts for one engine and two engine drives employing reduc-

tion gears and hydraulic couplings. Four-engine lay-outs may be arranged with the engines in tandem fore and aft of each of the pinions in a reduction unit similar to the one shown for the two engine installation. Other four-engine lay-outs use two engines connected to the pinions through hydraulic couplings, and two engines connected electrically with the motors driving the pinions in tandem with the direct drive engines.

COOLING SYSTEMS

All engines of recent construction are cooled by circulating fresh water through the engine water jackets. This water, leaving the engine warm, is passed through a heat interchanger of the shell and tube type, where it is cooled with sea water as the coolant. In some installations, the fresh water and the salt water pumps are driven by the engine, but it is a more common practice in large installations to use separate pumps which are electrically driven.

Some systems are fitted with heaters for warming the water in the fresh water system so that circulating it will heat the engine for easier starting. With other engines, the flow of cooling water may be varied to avoid heating the engine too rapidly.

After the main engines are secured the cooling water pump should be kept running at least 15 minutes to absorb the heat in the metal and cool it sufficiently to prevent boiling the water in the engine jackets. This is especially necessary if the engine is cooled with sea water because of the danger of forming scale in the water jackets.

Most modern marine installations have closed, or double-circuit cooling systems, wherein fresh water removes heat from the engine, and, in turn, is cooled by sea water in a heat exchanger. Open- or single-circuit systems, provide cooling by pumping sea water directly into the jackets. This latter type is not desirable because it permits scale, foreign matter, and marine growth to accumulate in the jackets, instead of in the more easily cleaned heat exchanger. Only clean, soft water should be used in a closed cooling system, since a small amount of scale or grease deposited in the jacket walls can cause serious local overheating.

The capacity of pumps, lines, etc., for a particular installation determines the rate of flow of cooling water through an engine and the proper selection of such equipment is normally a function of the engines designer. However, occasion may arise wherein an engineer officer may need to know how much cooling water must be put through the jackets to properly cool the unit. In such a case the following approximate method of determining rate of flow may be valuable.

The heat liberated during combustion in a Diesel engine usually is distributed:

	Percent
To useful work (brake thermal efficiency)	30–35
To exhaust gases	30–35
To cooling water	30–35
Radiation, lube oil, etc	0– 5

Cooling water temperature should be limited to a maximum of 160° F., in most cases, because local spots within the engine will have higher temperatures and will cause evaporation of the water; the steam so generated will vapor-lock portions of the jacket, thereby rendering adequate cooling impossible. On the other hand, jacket water temperatures should be kept above 145° F. to maintain good engine performance. Therefore, the closed-circuit coolant should enter the block at 145°–150° F., and leave the block at approximately 160° F. The problem, then, is to put water through fast enough to carry away approximately one-third of the heat liberated in all cylinders, for a limited cooling water temperature rise.

Example.—How many gallons of cooling water per minute must flow through the jackets of a 2,100-horsepower engine to properly cool it? Allow 10° F. water temperature rise.

$$\frac{2{,}100}{3}\times\frac{33{,}000}{778}=29{,}700 \text{ B. t. u./min. to cooling water}$$

1 B. t. u. raises 1 pound of water 1° F.; therefore, $\frac{29{,}700}{10}=2{,}970$ lbs. of water per minute will carry off 29,700 B. t. u. for a 10° F. water temperature increase. One gallon of fresh water weighs 8.33 pounds, so

$$\frac{2{,}970}{8.33}=356 \text{ gallons per minute are required}$$

OIL FILTERS

Wear in an engine is caused by metal-to-metal contact between moving parts, or through the action of abrasive particles larger in diameter than the minimum thickness of the oil films in the engine. To reduce wear from the latter cause, Diesel engines have filters fitted in the lubricating oil circulating line and in the fuel supply line.

Strainers and filters are installed in Diesel engines for two reasons:

(1) They remove from the lubricating oil abrasive materials which tend to increase wear.

(2) They remove from the fuel oil foreign substances which would clog up the injector nozzles.

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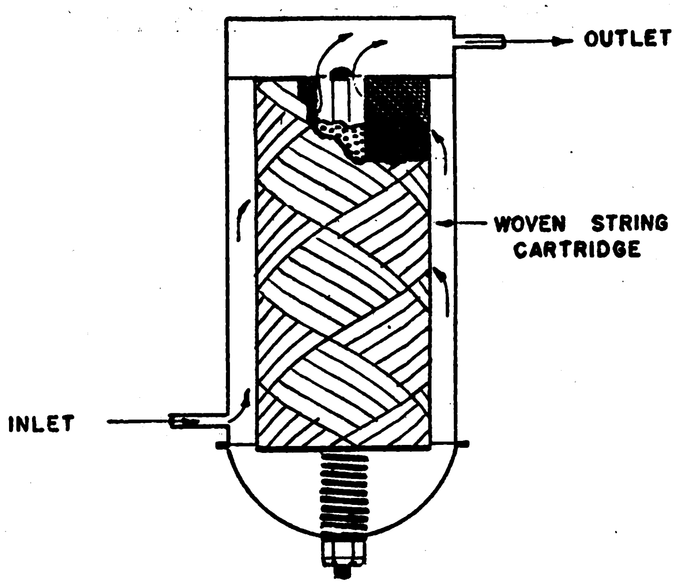
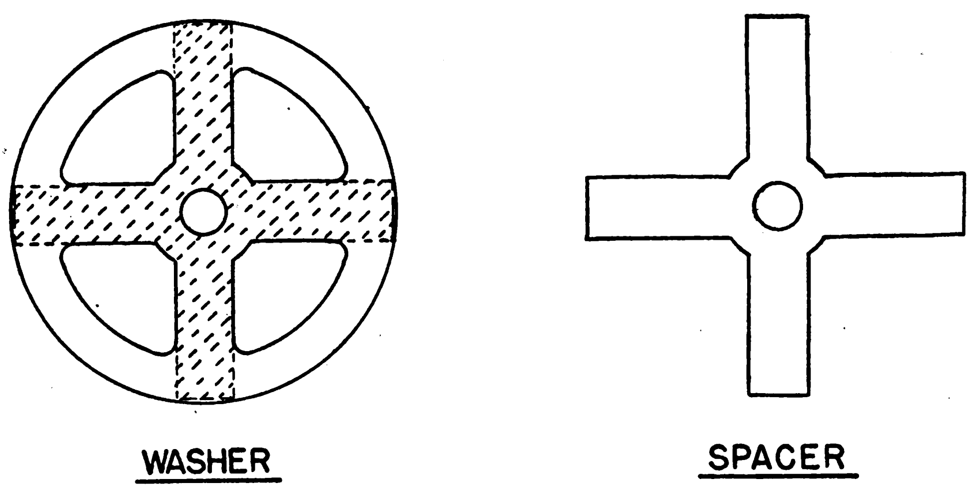


DIESEL FUEL FILTER

Figure 3.



ELEMENTS OF A METAL EDGE FILTER

Figure 4.

Strainers are devices which, by means of metal disc or wire mesh elements, separate the solid particles from the oil at the surface of the element. They are usually installed immediately in front of the filters in the oil line in order that they may remove the larger particles. Some metal edge strainers may be cleaned by rotating the knife edges past a cleaner blade, causing the sludge to drop into the sump. Other strainers are designed so that the entire strainer element may be removed and washed off. Metal edge strainers with openings of from 0.001 to 0.003 inch have been approved for use on Navy engines.

Filters employ an absorbent element to remove solids from the oil. This element may be made of a substance such as cellulose, cotton yarn, cotton waste, paper discs, or mineral wool. The path of the oil may be from the center of the element toward each end, from the bottom to the top of the cartridge, from the top to the bottom from the outside to an inside center tube, or from the inside out, depending upon the manufacturer. Due to the compactness of the filtering element, filters are capable of removing very fine particles from oil. Filters approved for Navy use must be of a replaceable element type.

A strainer is usually installed immediately in front of the filter in the oil line in order that it may remove the larger particles. However, in the lubricating oil system of a dry sump engine, the filter may be installed in the scavenge pump discharge line and the strainer in the pressure pump discharge line. Filters are sometimes installed in a by-pass line, which bleeds a certain amount of oil from the main pressure line and returns it to the sump. Filters and strainers should always be installed in the line ahead of the cooler, since best results are obtained when the oil is hot.

Duplex filters having a two-way switchover valve are recommended for fuel systems in order that one element of the filter may be cleaned while the engine is in operation. Strainers in fuel systems should also be duplex. In lubricating oil systems, a manually operated by-pass should be installed around the filter and another around the strainer so that the oil may be temporarily by-passed if it becomes necessary to replace or clean an element when the engine is in operation.

OIL COOLERS

All marine Diesel engines have coolers installed in the lubricating oil circulating system for controlling the temperature of the oil delivered to the bearings and to make it feasible to use the oil as a coolant in the engine. The coolers are usually of the shell and tube type, and the coolant is sea water. A positive pressure should be maintained at all times on the oil side of the cooler to prevent the entry of salt water

into the lubricating oil system. The sea water side of the cooler should be drained when the oil circulating pumps are secured.

Lubricating oil for the reduction gears and for hydraulic couplings is usually circulated in a system separate from the engine system. The equipment is similar.

ALARMS

Alarms are usually installed to warn the engine room watch if the lubricating oil pressure in the engine or the reduction gear systems falls below a safe operating value. This alarm also sounds on the bridge.

ENGINE GOVERNORS AND OVERSPEED TRIPS

Diesel engines, whether used for main drive engines or as auxiliaries, are usually fitted with governors. Various types of governors are used. Most of the units are of the flyball type, with or without follow-up mechanisms to improve regulations characteristics as may be required by load conditions. Isochronous governors are required for Diesel engines driving A. C. generators. The construction of these governors follows the principles used for turbines performing similar service.

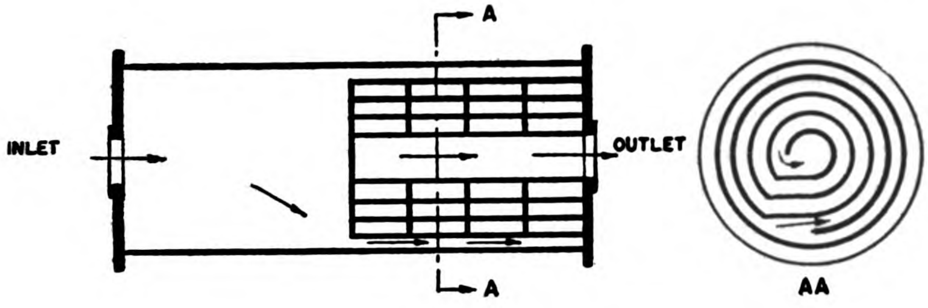
Main drive engines are fitted with overspeed trips in addition to a standard type of governor. One type of overspeed trip uses a centrifugal oil pump driven by the engine. When excessive speed is reached, oil pressure developed by the pump acts on a hydraulic piston which lifts the injection valve rocker arm from its cam and prevents further delivery of fuel to the engine.

Engineer officers should thoroughly familiarize themselves with all governors mounted on engines under their supervision. Descriptive material furnished by the engine manufacturer will be invaluable in this regard.

AIR COMPRESSORS

Compressed air for starting Diesel engines is maintained at about 250 to 400 pounds per square inch gage for solid injection engines, and at 500 to 750 pounds per square inch gage for air-injection engines. In most installations this air is compressed by an electric-motor-driven two-stage or three-stage compressor. These compressors are built with unloading valves so that they automatically cease pumping when full pressure is developed in the air starting bottle. Because of the uncertain demand for air during maneuvering, it is common practice to run the air compressor continuously while operating in restricted waters.

The manufacturer's recommendation for the lubrication of air compressors must be strictly followed. If any appreciable quantity



MAXIM SILENCER
Figure 5.

of low-flash-point lube oil found it's way into a compressor cylinder a disastrous explosion might easily occur, since all the essentials for Diesel-type combustion would be present.

MUFFLERS

In order to reduce the noise of the engine exhaust, a muffler is fitted some place between the engine manifold and the outlet to the atmosphere. In its simplest form it consists of a chamber in which reexpansion of the gas occurs and the direction of gas flow is changed. A Maxim silencer is shown in figure 5. The silencing unit consists of baffles which form spiral passageways. The gases enter these units at the outer ends of the spirals and flow inward to the central outlet. The baffles act to reduce pressure fluctuations and serve as an acoustical labyrinth to absorb sound generated up-stream of the muffler.

SUPERCHARGING

Most two-cycle, and many four-cycle, marine Diesels of modern construction are supercharged. Centrifugal blowers or positive displacement rotary pumps are usually used for compressing the air for supercharging. Many installations are gear-driven from the engine, and the power required to drive the pump or blowers is at expense of engine power supplied to the shaft. Another type is driven by a turbine motivated by exhaust gases. In this latter type, much of the energy which otherwise would be wasted up the stack is utilized to drive the turbocharger, as this type is usually called.

Practically all of the power required to drive the turbocharger is power which ordinarily would be wasted, so this type can be said to present no drain on the shaft horsepower (b. h. p.) developed by the engine.

In practice, the speeds, efficiency, and general performance of a Diesel engine may be limited by cylinder temperatures attained, or by the amount of oxygen available for combustion (volumetric efficiency). Supercharging improves both conditions. The maximum temperature in the cylinder is a function of the weight of air to weight of fuel ratio. Therefore, if intake air is supplied at, for instance, 5 p. s. i. gage pressure (approx. 20 p. s. i. absolute), one-third more oxygen will be available for combustion for the same maximum temperature, and one-third more fuel can be burned efficiently.

The supercharging pressure increment is added to the m. e. p., thus boosting power output.

Supercharging blows air right through the cylinders, thus aiding scavenging. Furthermore the blow-through cools cylinder "hot spots," and volumetric efficiency is further improved. All of these improvements lead to higher engine speeds, hence to greater power output.

Ordinarily, supercharging is arranged to increase the engines output by about 50 percent more than that of the unsupercharged engine. Theoretically, supercharging could be carried far beyond this point, but practical considerations of cost, size, weight, and mechanical difficulties usually make further development uneconomical with present materials and construction methods.

Operating personnel must carefully watch the exhaust gas temperature if a turbocharger is installed; turbine blades weaken rapidly above certain temperatures (usually about 1,000° F.–1,200° F.), and manufacturer's recommendations in this regard must be followed closely.

INDICATORS

Indicator cards may be taken for slow-speed Diesel engines with the same type of card-drawing indicator used for air compressors and reciprocating steam engines. Where its use is feasible, this type of indicator may be used to determine many details of engine performance.

The speed of modern Diesel engines in geared installations is too great for the usual type of indicators. The inertia of the moving parts causes the instrument to draw inaccurate and erratic cards. High-speed indicators are available and are used as research tools. One type uses an electronic indicator with piezoelectric electric crystals or magnetic-striction-type pressure elements. A cathode ray tube is used with the system and the pressure-volume trace is made on the screen of this tube.

A type of indicator known as the Premax pressure indicator is now being furnished to Diesel powered ships of the Navy. This instrument does not produce an indicator card, but does provide a method of measuring compression pressure and maximum firing pressure.

The complete outfit consists of the indicator, a flasher, and a connecting cord. The indicator contains a small piston which is spring loaded with a calibrated spring. This spring holds the piston against a stop unless the gas pressure in the indicator cylinder, which is connected to the engine cylinder, exceeds the value for which the spring is set. When this occurs the piston rises slightly, actuating a switch which causes a neon lamp in the flasher unit to blink. The spring tension is adjusted by turning a sleeve on the outside of the indicator. This sleeve is graduated after the fashion of a micrometer and shows the pressure to within 2 pounds per square inch. The spring tension is adjusted until a slight increase will prevent the lamp from flashing, while a slight decrease will cause the lamp to resume flashing.

The indicator is connected to an indicator cock communicating with the engine cylinder. With the engine running at normal speed, the pressure developed in each cylinder when firing, and also without fuel injection, is determined. The firing pressure data assist in balancing the load on the individual engine cylinders. The compression pressure data give warning of ring failures or wear, and assist in localizing trouble if the engine fires erratically.

Another indicator now being furnished to Diesel powered vessels is the Kiene maximum pressure indicator. This instrument can be used to measure compression pressure and maximum firing pressure.

The instrument consists of a pressure chamber, check valve, and pressure gauge. It is attached to the indicator cock by a threaded wing nut arrangement. The gases from the cylinder enter the pressure chamber through the check valve, the pressure inside this chamber being measured directly by the pressure gauge. Special instructions for cleaning and maintaining these indicators are supplied with the instruments.

PYROMETERS

Nearly all marine Diesel engines, whether used for ship propulsion or for auxiliary service, are fitted with exhaust pyrometers. These pyrometers make it possible to check the exhaust temperature in each branch of the exhaust manifold as the hot exhaust products are ejected from the engine. By maintaining a log of these temperatures, changes in the operation of injectors are easily detected.

The temperature of the exhaust is a good indication of the amount of load being carried by the cylinder in question. A high temperature indicates a heavy fuel injection. The temperature in each of the several branches of an engine exhaust manifold should normally be about equal. Abnormalities should be checked by examining the fuel injectors; 50° F. is a significant temperature variation between cylinders in most engines.

Chapter IX. OPERATION

1. STARTING

ROUTINE STARTING

The following items include only those to be made at each engine start:

(*a*) Check position of all valves in the fuel, lubricating oil, and cooling systems.

(*b*) Start motor driven circulating water pumps, check gage pressures, and check piping and fittings for leaks, especially cylinder liner packing glands and fittings inside crankcase. Verify the flow of water through all cooling spaces, checking vent valves where installed. See that the system is full on engines with closed fresh water cooling systems.

(*c*) Fill lubricating oil sumps to proper level. If practicable, take sample from bottom of each sump. Pump away any water that is present and then run all the oil through the purifier.

(*d*) Check the oil level in the governor. Do not fill the governor above the proper level.

(*e*) Start detached lubricating oil pump and check oil supply to various essential parts of the engine. Check gage pressures. Examine oil lines for leaks.

(*f*) See that the fuel gravity tanks are filled, and that water and sediment are drawn off.

(*g*) See that the mechanical lubricators are filled and operate them by hand to check the flow through all outlets.

(*h*) Check grease cups and fill if necessary. Give each cup one or two turns as required. Lubricate by hand all rubbing parts that are not pressure lubricated.

(*i*) Blow out circulating water sea chests, and see that circulating water and lubricating oil strainers are clean.

(*j*) Rotate fuel and lubricating oil filter scraping blades, if furnished, and check pressure drop across the filters. Clean all screen or bag-type filters if required. Replace cartridge-type elements when necessary.

(*k*) See that the starting air flasks are filled to the correct pressure, with all valves in the stop position.

(*l*) Open all indicator cocks, cylinder test valves, or hand-operated relief valves.

(*m*) Be sure that the jacking gear is not engaged.

(*n*) On air-injection engines, open the air-compressor throttle. Close drains and vents, except the drain from the flasks. Open valves to the spray air flasks. See that all gage cocks are clear and secured in the open position and that all valves and other cocks are in the running position. Blow through the spray air line. Test the spray air-relief valves. Fill the spray air flasks to their designated pressure. Test the tightness of the spray valves by opening the indicator cocks with the starting lever in the stop position. If the exhaust and inlet air valves have not been tested recently, they may be tested with spray air pressure, taking care not to admit sufficient air to turn the engine over. Slowly open the valve on the spray air flasks connecting them to the air compressor and see that the air lines, high-pressure air coolers, and separators are tight. When suitable connections are available, the valve in the drain manifold may be cracked to admit air into the air coolers and separators of low and intermediate stages to test them for leaks. The corresponding gages must be watched so that ordinary running pressures will at no time be exceeded. The spray air lines may be tested by opening the valve on the starting air flask.

(*o*) Prime the fuel lines up to the injectors or spray valves, and bleed the system of air.

(*p*) Put all cocks and valves in the running position.

(*q*) Close indicator cocks or cylinder test valves.

(*r*) Set the governor control on engines with manually controlled governor adjustments, to the starting speed recommended by the engine manufacturer.

STARTING BY AIR, OR AFTER SURFACING

Immediately before starting by air, or when starting after surfacing on a submarine:

(*a*) Open the circulating water outboard discharge valve and, after a few minutes, close the air vents and drainage valves on the mufflers and evaporators.

(*b*) Jack the engine over by jacking gear at least one complete revolution with the indicator cocks open and then turn over slowly by air for a few revolutions, observing carefully for any indication of water in the cylinders which will show as spray at the indicator cocks. **The cylinders must be free of water before attempting to start the engine.** If water persists and is obviously not residual water from outboard exhaust valve leakage, the source of the leakage must be traced down and corrected.

(*c*) See that the starting valves are in the stop position.

(*d*) Open the inboard exhaust valve on submarines and close the drains from exhaust piping and manifold.

(*e*) Engage the clutch, if one is installed.

(*f*) Close vent cocks and valves on the starting and on the spray air lines.

(*g*) Close indicator cocks or cylinder test valves.

(*h*) Open the air compressor throttle on air injection engines.

(*i*) Place all valves and cocks on fuel, lubricating oil, and circulating water systems, in their running positions.

(*j*) If engines are reversible, set the reversing gear in the proper position.

(*k*) Keep the motor-driven lubricatng oil and circulating water pumps running until the engine and the service pumps take up the load.

(*l*) Slowly open the valves to the air flasks. The spray-air pressure on air injection engines should be at least 650 p. s. i. and may be built up from the starting or ship flasks, or by turning the engine over by motor until the necessary pressure is shown.

(*m*) Open the outboard exhaust valve as the engine starts to turn over.

CHOICE OF STARTING METHOD

Starting by air should be the normal practice with installations having an adequate and habitually dependable starting air system. Installations having no starting air system, or in which starting failures are frequent, may be started with electrical power. Starting by electrical power, if possible to do so, is always recommended when starting up the engine for the first time after a long period of idleness or when installing major parts, as it affords, due to slower acceleration, a better opportunity to observe carefully whether everything on the engine is in working order.

PROCEDURE WHEN STARTING BY ELECTRICAL POWER

In general, the procedure outlined for air starting will be carried out when starting by electrical power, with the following additional instructions:

(*a*) The fuel control must be set so that no fuel is admitted to the injectors or spray valves until the engine is up to the necessary speed and ready for firing.

(*b*) The starting levers must be in the stop position.

(*c*) The reversing gear, if provided, must be set in the same direction of rotation as the motor.

(*d*) The indicator cocks should be open.

(*e*) The engine must be started as slowly as possible and the speed increased very slowly. Observe the precautions for determining water in the cylinders.

(*f*) The starting levers are not to be shifted to running position until the engine is turning at the proper starting speed.

(*g*) Set the governor speed, on engines having adjustable governor control, to the speed recommended by the manufacturer for starting.

(*h*) Fuel must be admitted gradually and, when the engine fires and takes the load, electrical starting power should be cut out.

(*i*) Care should be exercised to prevent placing the controller in the reverse position when bringing it from the starting to the neutral position.

2. RUNNING

MAINTENANCE OF NORMAL PRESSURES AND TEMPERATURES

Observe the various gages, thermometers, tachometer, and water, fuel, and lubricating-oil levels carefully while starting, and at all times while running. If the pressures for lubricating oil, circulating water, and intermediate stages of the air compressor on air-injection engines, do not commence to rise immediately, the engine must be stopped and reported "out-of-order" until the cause is found and remedied. The normal pressures and temperatures to be maintained are issued with the instructions for each installation and should be carefully followed.

CRITICAL SPEEDS

Engines must not be operated at critical speeds as the resulting vibration will cause overstressing and eventual breakage of shafting, heating of bearings, and other serious damage. Critical speeds must be passed through as quickly as possible when changing engine speed. Detailed information relative to critical speeds will be issued with each installation.

TACHOMETERS TO SHOW CRITICAL SPEED RANGES

Tachometers will be marked to show ranges of critical speeds in which it is forbidden to operate except in an emergency. The tachometers are likely to get out of adjustment, however, and must be checked frequently with mechanical counters. Proper corrections to the tachometer indication must be made to insure that the engines are not operated at critical speeds because of erroneous readings.

BUILDING UP LOAD AFTER COLD STARTING

The load must be built up as gradually as the situation will permit when starting a cold engine. Follow the time intervals recommended by the builder when conditions of operation will permit.

OPERATION AT LOW LOAD

Do not operate the engine for any length of time at less than one-third load as combustion at low loads is incomplete and partially burned fuel and lubricating oil may foul the valves and cause heavy carbon deposits in the combustion chamber and exhaust system, also lubrication of the cylinder walls and pistons in engines not fitted with force feed cylinder lubricators is sometimes inadequate at low speeds and during low temperature operation.

ENGINE LOAD AND SPEED

Experience has shown that for minimum maintenance and engine derangements propelling engines should not be operated continuously in excess of 80 percent of the maximum rated load at 90 percent rated speed. Diesel engines are usually most efficient at approximately 75 percent b. m. e. p., and in general it is preferable to operate with a speed not lower than 75 percent of the rated speed and with the load at 70 to 80 percent of the rated b. m. e. p. **On a multiple engine installation, it is always preferable to operate a smaller number of engines on the line at cruising power than a large number at light loads or low speeds.** Neither the maximum allowable b. m. e. p. nor the rated r. p. m. sholld be exceeded.

OVERLOADING THE ENGINE

Reduce the load if there is any indication of overloading any cylinder and stop the engine at any loss of lubrication or cooling. If the exhaust temperatures are high, exhaust smoky, and the engine generally overheated, the engine is probably overloaded and the load should be reduced immediately.

GOVERNOR ACTION DURING STARTING

In actually starting an engine, after starting preparations and inspections have been made, the governor control should be set for recommended starting speed. The main air-starting valve should then be turned full on. It is important that this valve be open—not merely cracked—so that a full charge of air will be admitted to the cylinders, both to crank the engine and to sustain combustion during first few power strokes. The engine should fire during the first or second revolution, and the main starting air valve should be closed promptly.

The engine will accelerate rapidly until the governor pulls the fuel pump racks in, and holds the engine to the speed for which the governor has been set. Due to the rapid acceleration of a Diesel engine, the governor must take control of the engine within a very few seconds.

If the governor fails to take hold, the engine should be stopped immediately, or serious damage due to excessive speed will result. The faulty condition in the governor must be remedied before the engine can be started with safety. Hand throttling of an engine operating without load is not good practice.

TEST EMERGENCY TRIPS PERIODICALLY

Most Diesels are equipped with a device which directly or indirectly can render the fuel pumps inoperative. Such devices are provided to stop the engine quickly in an emergency.

Since the emergency trips are not used in normal operation, it is good practice to stop the engine several times a week, on routine stops, by means of the emergency trip. In this way the operating personnel can assure themselves that the device functions properly, and will not fail in an emergency.

CIRCULATING WATER AND LUBRICATING OIL TEMPERATURES

Circulating water should be at very nearly the same temperature for the discharge from each cylinder. Circulating fresh-water discharge temperatures should be not less than 140° F., to keep the engine at an efficient operating temperature and should approach the maximum allowed by the manufacturer. Circulating salt-water discharge temperatures should never exceed 130° F., as higher temperatures will cause deposits of salt and other solids in the jackets or cooling spaces, and should not be less than 100° F. for efficient engine operation. The water flow through the lubricating air cooler should be regulated to give a lubricating oil discharge temperature from the cooler as high as practicable, but in accordance with the maximum limit specified in the engine builder's instructions.

LUBRICATING OIL SUPPLY AND PURIFICATION

Watch that the mechanical oilers are feeding and are not airbound. Check the oil level in the sump tanks from time to time and keep them approximately three-fourths full. Keep the lubricating oil purifiers running at all times when one or more engines are in operation and for a sufficient length of time after shutting down to keep the oil clean. Turn the filter cleaning handles frequently.

INTERCOOLER DISCHARGE TEMPERATURES

Air discharged from an engine air compressor cooler must be as cold as possible and never higher than 140° F. The cooler tubes need cleaning if increased circulating water does not lower the air temperature.

LOGS

Keep complete logs of all operation, noting any unusual noise or occurrence. These logs will be of value in determining causes of derangements.

3. STOPPING AND SECURING

All personnel must be thoroughly familiar with the various details of stopping any specific engine. The manufacturer's instructions will list each step and should be followed in detail.

(*a*) Operate the motor-driven circulating water and lubricating oil pumps while the engine is cooling down.

(*b*) *Draining exhaust lines.*—Open the drain cocks on exhaust line and headers. A schedule of periodic draining of exhaust lines and headers sufficient to keep the gas passages clear of water must be established.

(*c*) Leave an adequate number of indicator cocks, cylinder test valves or hand operated cylinder relief valves open to indicate the presence of water. See that the pressure is off the spray air, and starting air lines. Possibility of serious accident exists if high presure air is left on.

(*d*) Close all valves. Allow the engine to cool and then drain the fresh water when freezing temperatures prevail if an antifreeze solution is not being used.

(*e*) Clean the engine and the floor plates thoroughly, and see that the bilges are dry. Correct any troubles discovered, regardless of how minor they may appear to be.

SUBMARINES ABOUT TO SUBMERGE

On submarines about to submerge:

(*a*) Close the outboard exhaust valve as soon as the engine stops rolling but not before. A one second lag but no time lead is permissible to keep from flooding the engine and avoid building up excessive pressures in the exhaust headers.

(*b*) Shut the inboard exhaust valve.

(*c*) Open the vent and the drain valves to the mufflers.

(*d*) Stop the circulating salt-water pump and close the sea valves. The motor-driven circulating fresh water and lubricating oil pumps should be operated until the engine cools down.

SUBMARINES WHILE SUBMERGING

On submarines while submerging:

(1) Observe the drains from the exhaust lines to make sure that the overboard exhaust valves are tight. These drains must be plainly visible. If leaks are found, start the bilge pumps in time to prevent water from reaching the electrical machinery or wiring while on an uneven keel.

(2) Watch circulating water gages to make sure that sea valves are tight.

Chapter X. TESTS AND INSPECTIONS

1. INSPECTIONS

Many of the inspections itemized below may not be necessary at the time of each period of operation except immediately following an overhaul or a long period of idleness. The engineer officer must decide which are necessary, depending on the condition of the engine, the time it has been idle, and the attention it has been given during the idle period.

BEFORE STARTING THE ENGINE

The following checks should be made before starting the engine:

(*a*) Check the fuel pump adjustments.

(*b*) Check the chatter and spray of the fuel injection valves on solid injection engines, and clean or replace any which do not pop at the manufacturer's recommended pressure or that do not chatter properly.

(*c*) Test spray valves on air injection engines. Lubricate valve spindles and test for freedom of motion. Grind in or replace if necessary. See that nozzle holes are open, clean, and of proper size. Check the fuel timing.

(*d*) Test all tappet valves in the cylinder head by hand for free movement. Lubricate with penetrating oil or kerosene, if stiff, and turn the valve on its seat or work it up and down until it moves freely. Clean any valves which cannot be made to move freely.

(*e*) Check clearances of the valve operating gear. Adjust valve lash take-up mechanism.

(*f*) Take out, clean, and reinstall engine air compressor valves.

(*g*) Keep the air compressor intercoolers, starting air flasks, and spray air flasks properly drained.

(*h*) Examine the condition of the seats, and the movement of the valves, on indicator cocks and hand-operated pressure relief or test valves.

(*i*) Examine zincs in coolers, and clean or replace them if necessary.

(*j*) Examine all bearings for excessive clearance, indications of wiping, and improper installation.

(*k*) Test the lubricating oil for the presence of any dilution before getting underway, and once during each watch while the engine is in operation. There is possibility of fuel or water getting into the system from leaking fittings, cooler tubes, gaskets, etc., and may so dilute the oil as to cause serious damage. Extensive dilution may be revealed by

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lack of necessity for adding the usual amount of make-up oil or by a rise in the sump level. Dilution with Diesel fuel may be determined by measurement of the viscosity. Normally, 5-percent dilution with fuel oil warrants rejection of the lubricating oil, although it may be used with dilution up to 10 percent in an emergency. Keep the filters and strainers clean and in good condition at all times.

(*l*) Check foundation bolts for tightness and see that all piping connections are properly made up. Make a final check of all clearances and adjustments.

(*m*) Examine all parts of the engine, particularly the crankcase and the working parts, for rags, tools, braces, or any other obstruction which might interfere with operation, and for improperly secured nuts, cap screws, and cotter pins.

(*n*) Using the detached pump, put pressure on fuel system and inspect all lines for leaks, especially where such leaks may allow dilution of lubrication oil.

(*o*) Check water level in the fresh water tank and strength of corrosion control solution if used.

(*p*) Open the cylinder test valves and turn the engine over with the jacking gear to make sure that there are no obstructions.

(*q*) Disengage the jacking gear.

(*r*) Latch the throttle in the closed position, and with cylinder test valves open, open and then quickly close the air starting valve. If no water or fuel is sprayed from the cylinder test valves, the starting air impulses are repeated until all pistons in turn have been brought to top center. If an excessive amount of water is sprayed from a relief valve, the leak should be located and corrected before the engine is started. If fuel is sprayed from the relief valve the injector should be replaced.

DURING OPERATION

The following checks should be made during operation of the engine:

(*a*) Watch all pressures, temperatures, speed, and load at all times during operation. See that the oil in the sump tanks is kept at the proper level.

(*b*) Watch all bearings and other moving parts very carefully and feel them from time to time when it can be done with safety. All rubbing or bearing parts that are not connected to the forced lubricating system must be oiled by hand every hour, and grease cups must be kept filled and turned down as required.

(*c*) Observe the action of the valve operating gear. Sticking valves may be detected by change in the clearance or in the take-up cam location.

(*d*) Check the outboard exhaust at least once a watch. The exhaust should be almost invisible at all loads. A dark exhaust indicates the engine is overloaded or out of adjustment.

(*e*) The engine must be kept properly cleaned at all times. Oil from leaks may result in a fire; dirt or grit may result in scratched or gouged bearings, or plugged pipes or fittings, and in loss of lubrication, causing a hot or seized part.

(*f*) Check pumping action of the fuel injector needle valve on solid injection engines, by pressing on the test needles, in injectors thus fitted.

(*g*) Spray valve spindles on air injection engines should be given a couple of turns every four hours, where provision is made to permit this, to prevent sticking of the spindle.

(*h*) Check starting air valves for leaks. The air line to the cylinder will be hot if the spring-loaded valve in the cylinder is leaking. Leaking cylinder relief valves may be detected as they become hot. A slight pulsating pressure may be felt at indicator cocks or cylinder test valves with bad seats.

(*i*) Check the water level in the expansion tank. Loss of cooling water or a surging water level may indicate a cracked liner or cylinder head.

(*j*) Investigate any unusual sound or noise and any irregular operation and take steps to remedy the trouble.

ROUTINE TESTS AND INSPECTIONS

In order to keep the engines in reliable running condition, it is necessary to dismount certain parts periodically for examination, cleaning, grinding-in, etc. The interval of time for this routine work is issued with the individual instructions for each installation. The following items should be checked in addition to any included in the manufacturer's instruction books or in fleet instructions:

(*a*) Test lubricating oil for the presence of water before getting under way and once during each watch. If any water is found, check oil coolers and engine circulating water system for leaks.

(*b*) Observe the exhaust at least once each watch.

(*c*) Once every 24 hours while under way, check the lubricating oil low-pressure alarm and circulating water high-temperature alarm.

(*d*) Every 24 hours of engine operation, check the lubricating oil for fuel dilution.

(*e*) Atomizer holes should be inspected frequently, and should be cleaned when dirty or partly plugged.

(*f*) Cams and air-injection spray valves should be inspected once a week.

(*g*) Jack an idle engine over several times weekly, leaving it in a different position each time, in order to prevent staining and possible corrosion of the cylinder walls.

(*h*) Air compressor valves should be removed and inspected at least once for every 150 service hours.

(*i*) Every 200 hours take a sample of lubricating oil and send it to a navy yard laboratory or to the Engineering Experiment Station at Annapolis for analysis.

(*j*) Twice a month test the operation of the overspeed governor by suddenly increasing the engine speed with the hand throttle.

(*k*) Check engine thrust bearing clearance every 400 hours.

(*l*) Valve timing adjustments and settings should be tested once each month.

(*m*) All valves should be inspected and reseated adequately for their service about every 500 hours.

(*n*) To keep spare governors of the Woodward type in running condition, they should be interchanged every 3 months with the ones in service. Check all governor linkage for wear or play.

(*o*) Relief valves should be tested every 3 months.

(*p*) Zincs should be inspected at least once every 3 months, and oftener if the rate of deterioration warrants.

(*q*) Air-compressor pistons should be withdrawn and cleaned at least once in 6 months.

(*r*) Drain the fresh-water circulating system every 6 months and flush out all sediment which may have accumulated.

(*s*) Exhaust mufflers and exhaust lines should be inspected and cleaned at every navy yard overhaul.

(*t*) Fuel tanks and lines should be inspected and cleaned at every navy yard overhaul.

(*u*) Air flasks should be tested hydrostatically at pressures 50 percent in excess of working pressure at intervals not exceeding 5 years.

(*v*) Working cylinder and cylinder head circulating water passages are to be inspected for scale, and cleaned and tested hydrostatically at the manufacturer's recommended pressure at each overhaul period.

(*w*) Any part or type of parts in which there is known to be possibility of failure because of design or previous record of failure should be inspected as frequently as experience indicates to be necessary.

2. TEST PROCEDURES

DETERMINATION OF PERCENT DILUTION OF LUBRICATING OIL

The percentage dilution of lubricating oil by the Diesel fuel oil can be found by use of the viscosity blending chart shown in figure 41–47. A line is drawn between the lubricating-oil viscosity marked

on the right vertical boundary line, and the Diesel fuel-oil viscosity marked on the left vertical boundary line. The intersection of this

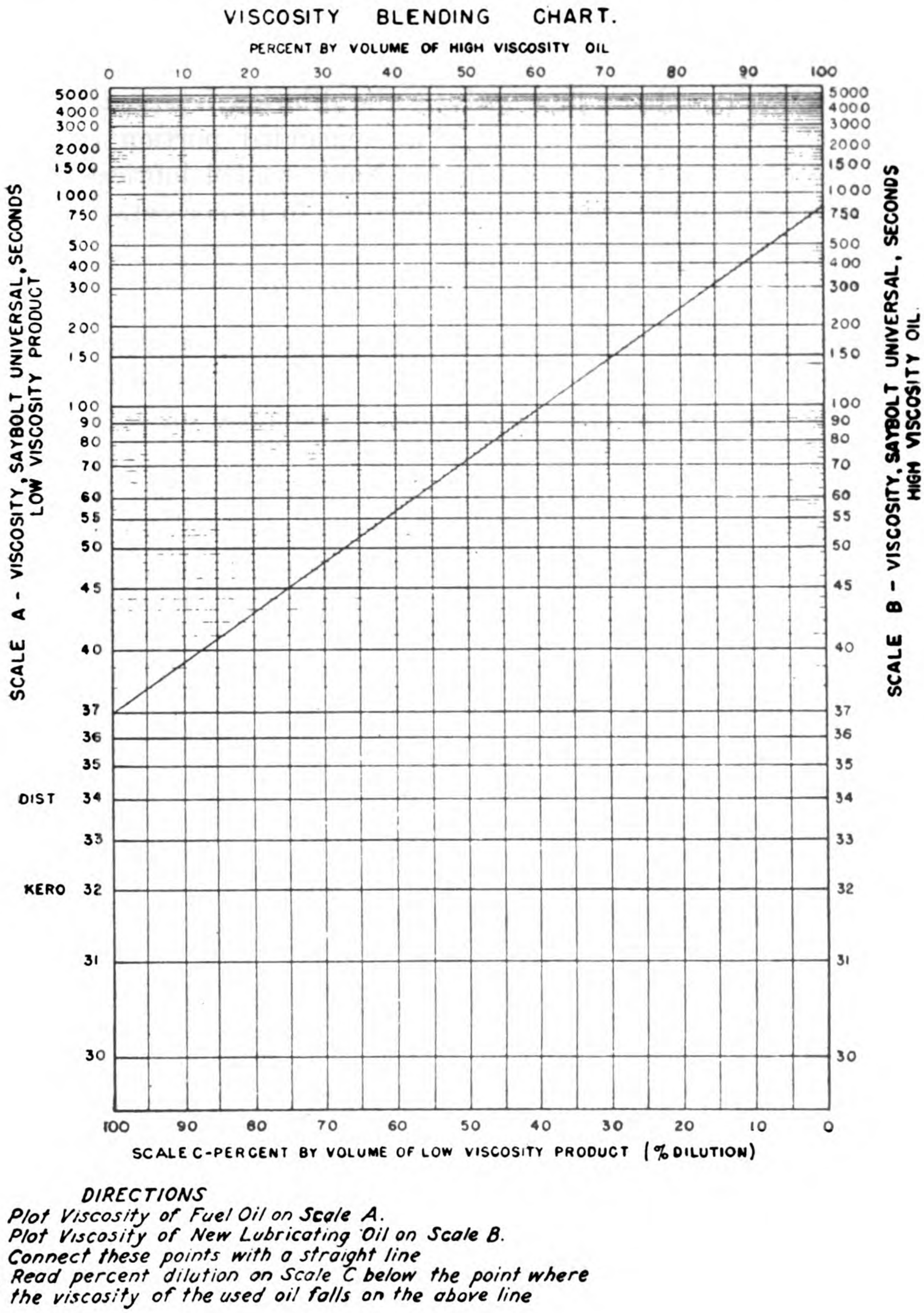

FIGURE 41-47

line with the viscosity line of the used lubricationg oil will show the percentage of Diesel oil (percent dilution) in the used lubricating oil.

Example

	Saybolt Universal viscosity seconds at 100° F.
New lubricating oil	800
Diesel fuel oil	37
Used lubricating oil	580

As shown on the chart, figure 41–48, the dilution is approximately 5 percent. Chart 41–48, which is an expanded portion of 41–47 with the dilution curves drawn in for Navy symbol lubricating oils may be more readily used for oils diluted up to 10 percent.

WATER IN LUBRICATING OIL

The presence of water in lubricating oil may be detected by testing with a commercial product known as Indicating Drierite. To make this test, place a 25-cc. sample of crankcase oil in a 100-cc. dry test tube. To this sample, add 75 cc. of a good grade of kerosene or clear (free from water) Diesel fuel oil. Shake the test tube until the contents are well mixed. Add one granule of Indicating Drierite to the mixture, shake well and allow to stand for about 5 minutes. If the granule is not discolored or only partially discolored, there is less than 1 percent of water present; if the granule turns white, more than 1 percent of water is present. Care should be taken to keep the Indicating Drierite dry until actually used for test. In removing a granule from the bottle a tweezer should be used to prevent change in color while transferring from the container to the test sample of lubricating oil. Indicating Drierite is carried in stock at all Navy Yards.

CORROSION CONTROL IN CLOSED COOLING SYSTEMS

(1) The use of corrosion inhibitors is mandatory in engines having steel frames in contact with the cooling water and is advisable in all engines with closed cooling systems. The method of treatment outlined in these instructions consists of the maintenance of the pH (alkalinity) of the water in the range in which corrosion is inhibited, and simultaneously maintaining sufficient phosphate in the water to prevent scale deposition on the heat-transfer surfaces. It has been found by test that, if trisodium phosphate issued for maintenance of pH, sufficient phosphate will be added thereby to react with any scale-forming salts in the make-up cooling water. This added phosphate, however, will not be sufficient to remove scale deposits already present in the cooling systems.

(2) *Method of test.*—The only test required for the control of the treatment consists of adding corrosion-control indicator, Navy Department Specification 51–I–3, to a sample of the cooling water.

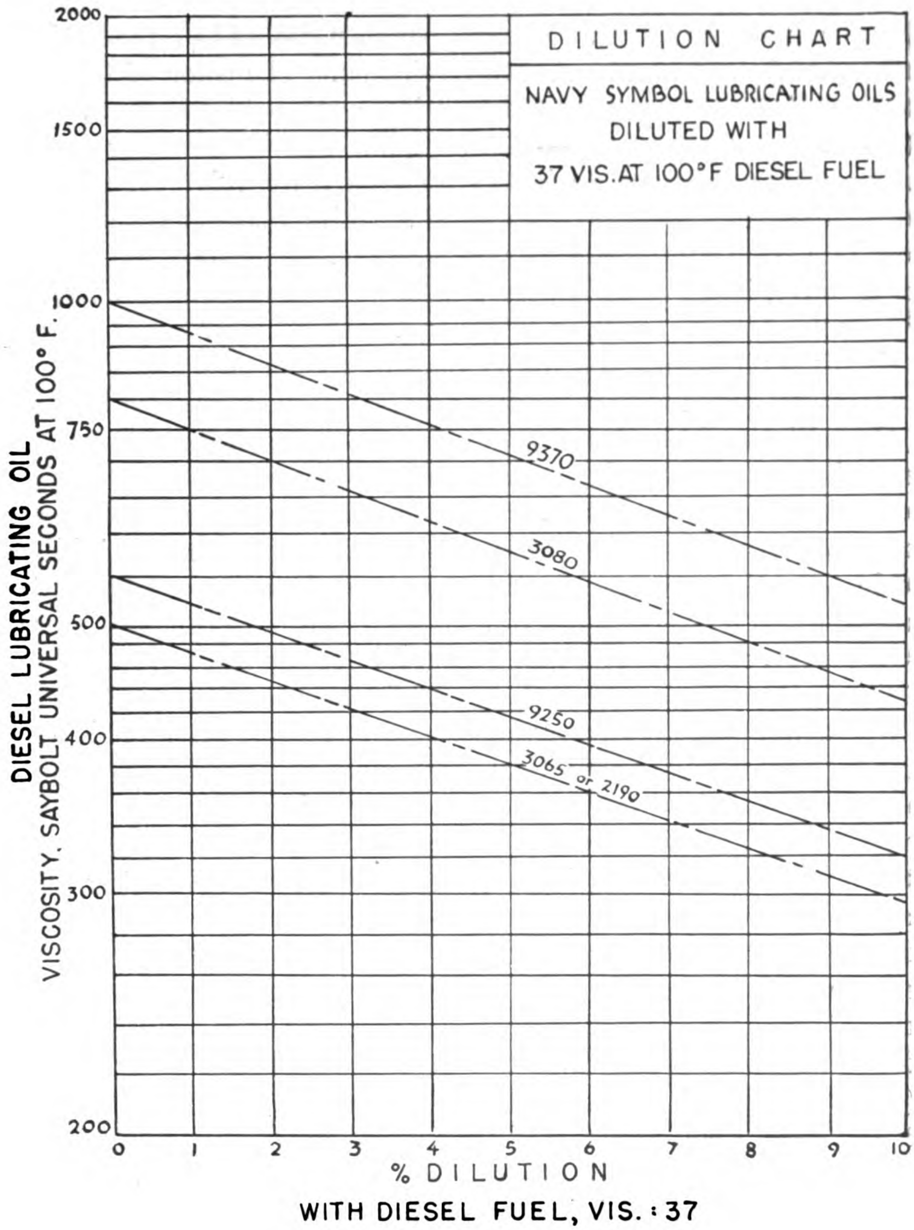

FIG. 41-48

The resulting color of the sample indicates the condition of the cooling water, as follows:

Yellow—insufficient treatment.
Red—satisfactory.
Purple—excess treatment.

A test is made by drawing a small sample of water from the cooling system. This sample should be cooled to 85° F. or below. About 10

milliliters should be transferred then to a test tube and one drop of indicator should be added. The stopper then should be replaced in the test tube and the test tube should be shaken. The color of the water should be observed in a good light against a white background.

(3) *Treating compound.*—The compound to be used for treating cooling water consists of a mixture of six parts by weight of trisodium phosphate and one part of cornstarch. Ships may obtain trisodium phosphate from supply bases or navy yards. Cornstarch may be obtained conveniently from galley stores. The small amount required may be mixed in small lots on shipboard. A lot consisting of 6 pounds of trisodium phosphate and the contents of a 1-pound package of cornstarch can be mixed with a paddle in an ordinary galvanized iron pail.

(4) *Initiating water treatment.*—As stated above, the treatment is a preventive treatment and not adequate for the removal of scale deposits already in the cooling system. Therefore, if the system is not new or is not known to be absolutely clean, it should be cleaned with inhibited acid in accordance with the instructions of the Bureau of Ships. The clean and well-flushed system should be filled with clean, fresh water. The capacity, or amount of water from the freshly filled system should be taken and tested as described above. If the water is suitable for use, the test sample should have a yellow color. In this case, add a standard dose of 1 ounce of treating compound per hundred gallons of cooling water. The dose of compound should be completely dissolved in warm (not hot) water and added to the circulating pump suction. After a period of circulation sufficient for thorough mixing of the treatment with the cooling water a second sample should be drawn and tested. If this sample shows a red color, the initial dose was adequate. If the color of the test sample remains yellow another standard dose should be added, allowed to mix with the cooling water, and this process repeated until a red color is obtained.

(5) *Standard dose.*—The standard dose of 1 ounce of treatment per 100 gallons of cooling water is approximately the amount necessary to increase the pH to the red range. **This dose must not be exceeded.** If a larger dose is added, it is possible that the pH of the cooling water will be increased from the unsatisfactory low pH of the yellow range to the equally unsatisfactory high pH of the purple range. If only standard doses are added, and are allowed to mix completely with the cooling water before test samples are drawn, the purple range never should be entered.

(6) *Maintenance.*—The water in each cooling system should be tested once each week. If the test sample gives a red color upon the addition of indicator, the condition should be recorded as satisfactory and no treatment need be added. If the test sample gives a yellow color, a standard dose of treating compound should be added. The

water then should be retested to insure that sufficient treatment has been added. If the test sample gives a purple color, too much treatment has been added. In this case, about one-fourth of the cooling water should be drained from the system and replaced with fresh water. If on retest the purple color persists, it probably will require less expenditure of fresh water to drain and refill the entire system rather than to continue with partial drainage and refillings. In any case, the red colors of the test sample must be restored.

3. DERANGEMENTS AND CASUALTIES

INSPECTIONS

A written record should be made of each derangement or casualty. This record should include a description of the part after failure and notes on any events in the history of the part or of similar parts which might afford a clue to the cause of failure. The inspection of the failed part should be made very carefully so that all details bearing on the failure will be noted.

MEASUREMENTS

Sufficient measurements should be taken of every new part to insure that it is within the tolerance shown on the approved drawings. No part should be used which is not in agreement with the plans. Complete measurements should be taken of all parts which are subject to wear and which may later be rejected because of wear. These parts should be measured at intervals during their life, when opportunity permits, without interfering with operation of the engine. Analysis of these interim measurements will permit determination of the time when a part should be renewed to avert approaching failure. Frequently, failed parts are not in condition to permit measurement. When possible, however, they should be completely measured to determine the condition at time of failure.

REPORTS

Existing regulations and instructions prescribe the routine reports which should be submitted for every casualty or derangement. When considered necessary by the Forces Afloat, special reports may be forwarded containing the history of the part involved, or of other parts which have failed in the same manner, a detailed description of the type and circumstances of the failure, any measurements available on the part, comments on the failure, and recommendations to prevent recurrence of the failure. Reports of this type will aid the Bureau in preventing failures due to improper design or materials.

REJECTION AND REUSE

Some parts are usable after being involved in a casualty. Experience must decide the possibility of reuse of a part in each case, based on previous knowledge of the part and on wear measurements. A general safe rule is that when there is any doubt, or if resulting failure would cause extensive damage to other parts, that the part approaching failure be rejected immediately. In most cases some parts of the assembly can be reused, even though extensive damage may have been sustained by other parts. Some worn parts, such as shafts or pins, may be built up by metal spray and finished by grinding, or by cleaning the worn section and rebushing the bearing to fit. Other parts, such as piston crossheads and cylinder liners, may be stoned smooth after light scoring, provided the parts are not grooved deeply enough to permit blow-by.

CAUSES OF FAILURE

Determine the cause of failure as accurately as possible, considering all factors and making tests where necessary. It is realized that there will be some failures because of design imperfections, despite the preliminary engineering research which went into the machinery. There will probably be more failures because of the material of an individual part. It is believed that many derangements interfering with scheduled engine operation can be attributed to personnel. Only a small part of these derangements will be from careless operation. The greatest number will occur from two causes: First, failure to recognize that a part is nearing the completion of its useful life and then renewing it before it can cause a derangement; and second, improper installation and running-in of new parts and lack of proper cleanliness at all times. Accurate records of wear measurements and derangement inspections will provide a knowledge of the useful life of most parts and minimize derangements from parts which should have been previously removed.

PREVENTION BY REPLACEMENT

Prevent engine failures by replacement before failure, close observation of engine operation, and analysis of wear data. Parts should not be renewed indiscriminately or needlessly, but as a result of sound judgment based on analysis of conditions and a thorough knowledge of the installation.

4. SAFETY PRECAUTIONS

GENERAL

In addition to the specific safety precautions, operation personnel must exercise good judgment and employ common sense to prevent damage to material and injury to personnel.

(*a*) *Damage to machinery* may be prevented by proper operation of the engines in accordance with instructions, by cleanliness in handling all parts of the engine, by a thorough knowledge of duties, and by a complete familiarity with all parts and functions of machinery.

(*b*) *Damage to ship* may be prevented by handling the machinery so that there will be no loss of power at inopportune times and so the engines will be ready for service in any emergency, and by preventing conditions likely to constitute fire or explosion hazards, such as improper ventilation of the crankcase or control of overboard valves.

(*c*) *Injury to personnel* may be prevented by a thorough knowledge of duties, by proper handling of tools and parts, by normal precautions around moving parts, by adequate guards at constantly exposed danger points, and by training to eliminate carelessness and thoughtlessness.

UNINTENTIONAL STARTING

Precautions against unintentional starting of the engine must be carefully carried out before commencing any overhaul of the engine. The main engine clutch must be disengaged. Before working on an engine connected to a propeller shaft by a hydraulic clutch, engage the jacking gear to avoid the danger of windage in the clutch turning the crankshaft, should main motors in a compound drive be started or the ship be under way.

RELIEF VALVES

(1) If one of the relief valves, either on the working cylinders or on the air compressors, blows several times, the engine shall be stopped immediately and the cause of the trouble determined and remedied.

(2) Relief valves shall never be locked closed, except in case of emergency.

(3) Pressure relief mechanisms are fitted on all inclosures where the ignition of oil vapor may possibly occur. Strict compliance with their designated dimensions, adjustments, etc., is essential.

FUEL

(1) Precautions should be taken to see that fuel is not pumped into the cylinder while testing valves or motoring the engine, as an excessive pressure may result when the cylinder fires.

(2) The fuel should be absolutely free from water and foreign

matter when it reaches the injector. The fuel must be centrifuged thoroughly before using and the filters must be kept clean and intact.

(3) Leaks of fuel into the lubricating oil system must be avoided to prevent dilution of the lubricating oil with consequent reduction in viscosity and lubricating properties.

WATER

(1) Under no circumstances shall a large amount of cold water be allowed to enter a hot engine suddenly. The rapid cooling may crack the cylinder liner and head or seize the piston.

(2) When the volume of circulating water cannot be increased and the temperatures are too high, stop the engine.

(3) In freezing weather all parts containing water and subject to freezing shall be carefully drained unless, as in the case of fresh-water-cooled engines, antifreeze solution is added.

AIR

(1) When the engines are stopped, all spray air and starting air lines must be vented. Serious accident may result if pressure is left on.

(2) When the main engines are running and it is required to ventilate inboard, batteries should never be charged to such a state as to gas freely, except in emergencies or in required engineering runs. It has been definitely determined that under these conditions the air to the engine is strongly contaminated with sulphuric acid vapors, which combine with the iron and carbon to form a ferrous sulphate, an extremely hard and gritty substance, destroying lubrication and scoring bearing and rubbing surfaces. Except in emergencies, finishing charges and overcharges should always be made under conditions permitting outboard battery ventilation.

(3) Air for combustion must be kept as clean as possible. Accordingly, ducts and passages must be kept clean.

CLEANLINESS

(1) Engines shall be kept clean at all times, and the accumulation of oil in the bilges or other pockets should be prevented.

(2) Care must be taken that water in the bilges does not reach electrical machinery or wiring, particularly when on an uneven keel.

(3) Cleanliness is one of the most important basic essentials in operation and maintenance of Diesel engines. Clean fuel, clean air, clean coolants, clean lubricants, and clean combustion must be maintained.

(4) Do not use burlap for wrapping journals or polished bearing surfaces, due to the acid in the burlap.

Chapter XI. INSTALLATION OF NEW PARTS

INSTALLATION OF NEW PARTS

The clearances of all new parts should be checked before operation. Examine the installation for proper tightening of all nuts and cap-screws, securing of cotter pins, lubrication of surfaces, and removal of tools, rags, and other foreign materials. Make all necessary adjustments or settings. Jack the engine over by hand for at least two complete revolutions before starting the engine.

"RUNNING IN" OF NEW PARTS

The initial operation of new equipment or parts aboard ship should be handled as carefully as was the first operation of the engine on the test floor. The motor-driven lubricating oil pump should be used to force oil through the engine for at least 5 minutes before starting and the engine should be examined to make certain that the oil flow reaches all parts. The engine should be jacked over by hand several times so that the new parts will be well lubricated. Have the clutch throwout, the tailshaft coupling disconnected, or the generator field switches out, depending on the type of installation, so that there will be no load on the engine.

PROCEDURE FOR RUN-IN OF NEW PARTS

Roll the engine slowly by compressed air or electrical power, with the throttle closed, for about 5 minutes, and then stop and examine the new parts, if practicable, for indications of hard bearing or excessive heat. Examine contiguous parts. See the manufacturer's instruction book for recommended procedures for running-in new parts. The general schedule for generator engines should be:

(*a*) A short period of no-load operation at reduced speed.

(*b*) Light-load operation at reduced speed.

(*c*) Half-load operation at about 80 percent speed.

(*d*) Half load at full speed.

(*e*) Three-quarter load at full speed.

(*f*) Full load at full speed. For direct propulsion engines the load should be increased to full load in one-quarter load increments.

(*g*) Inspect the parts before each increase in load or speed. Total running-in operation should be from 12 to 24 hours, depending on the loading of the part and the results of the interim inspections.

PROCEDURE AFTER OVERHAUL

After completing overhaul of engine or renewal of complete main unit, in direct drive installations with the motor on the propeller shaft, the engine should be turned over slowly with main motors for at least 4 hours. Cruising m. e. p. should not be applied until after at least 24 hours of operation, and full power m. e. p. not until after at least 40 hours of operation.

"RUNNING-IN" OF NEW PARTS OF UNIT

When a piston, liner, crankpin bearing, or other part affected by the load of one cylinder only, has been replaced by a new part or unit, the engine may be placed in service as soon as the first full-speed inspection has been made, with the throttle for that cylinder adjusted to the allowable load. The allowable total load for the engine must be decreased accordingly. Increase the throttle on the new cylinder as the schedule permits, until it is carrying its full load. For engines not equipped with individual cylinder throttles the run-in procedure should be the same as if the complete engine was overhauled insofar as the exigencies of the service permit. Continue operation at reduced load on any part that is not polishing properly. Remove for refitting any part that shows hard bearing or incipient galling and then rerun the complete schedule.

EXAMINATIONS DURING RUN-IN

The examination of other parts affected by the operation of the new part is very important. A new crankshaft main bearing affects the loading on all other bearings and each one should be examined, if practicable, because of possible effect on the shaft alignment or whip. The same is true of gears, where one defective tooth in one gear may affect every gear in the train. The proper running-in of new parts will greatly extend their life. Keep an accurate and complete log on the findings of every inspection of new parts. In many cases an inspection of the logs and analysis of the bearing marks on the new parts, will later help to determine the cause of subsequent failure.

Chapter XII. MAINTENANCE

1. GENERAL—SCHEDULES AND OVERHAUL PERIODS

MAINTENANCE SCHEDULES

Maintenance schedules for each engine shall be prepared by the engineer officer which will provide for the periodic inspection of all engine adjustments and wearing parts at such intervals as may be necessary to maintain the power plant at maximum capacity and reliability. The schedule would include the tests and inspections recommended by this manual, by manufacturer's instructions and by Bureau instructions as well as any others indicated by experience. These schedules should be arranged in the form of check-off lists which list the items to be inspected after definite periods of operation.

INSPECTION RECORDS

Complete records of each inspection shall be kept which will permit an accurate observation of progressive wear or failure and will enable replacement before actual failure occurs.

OVERHAUL PERIODS

It is impossible to lay down hard and fast rules for the time interval between inspections and overhauls, as conditions vary greatly with various types of installations and with the severity of operation. The interval of time for some routine work is established for some engines. In cases where service conditions interfere with the maintenance schedule, the routine must be carried out at the first opportunity afforded after the specified interval of time has elapsed. Responsibility rests with the commanding officer to make proper requests for overhaul periods as required.

PROGRESSIVE MAINTENANCE

When operations permit, it is desirable to perform the engine overhaul work on one or two cylinders at a time with the maintenance work distributed at regular intervals over the overhaul period. For instance, if the Bureau's instructions require that a 16-cylinder engine be overhauled at 1,500-hour intervals, 2 cylinders should be opened up progressively after approximately each 200 hours of

operation. By following a progressive maintenance program, less time will be required to do the same amount of work than would be required if the engines were completely disassembled, all maintenance work performed, and the engine again assembled. This is due to the fact that many other things interrupt the work on an engine that is completely disassembled. Men regularly assigned to do the maintenance work become thoroughly familiar with the engine and the results are much better than if the work is put off until a major overhaul necessitates the use of a large number of navy yard men who are unfamiliar with the engine. Also, the men have more opportunity to become familiar with, and check the condition of the engine and this will prevent serious failures that may occur where an attempt is made to run the engine for the full overhaul period without other than superficial maintenance.

LAYING UP ENGINES TEMPORARILY

(1) All engine parts should be covered with a thick coat of grease or with approved rust preventitive compound to prevent corrosion when an engine is not to be used for a considerable length of time. (2) In order to make certain that all bearing surfaces of the cylinders and pistons are well oiled, after cleaning, jack the engine over several times with oiling and lubricating systems in operation. Repeat this process once a month during lengthy shut-down periods.

DRAIN COOLING SYSTEMS

All water jackets, cooling chambers, etc., must be thoroughly drained and blown out one at a time, in freezing weather, using *low-pressure* air. Approved antifreeze solutions may be used to obviate the necessity of draining fresh water systems. Proper blowing out of the water can be accomplished only by closing off all cooling spaces and emptying them separately.

INSPECT ZINCS

Zincs should be inspected frequently enough to prevent deterioration sufficient to allow electrolytic action on the coolers. The purpose of the zinc is to permit the electrolysis caused by dissimilar metals in the presence of salt water to act on the zinc rather than on the iron and other metals of the cooler.

CARBON REMOVAL

Carbon should never be scraped from the bearing surfaces of the pistons, valves, or other moving parts, or from the piston ring

grooves, with a metallic scraper. It is advisable never to remove any piston ring from its groove unless it is absolutely necessary or the ring is to be rejected. The rings may be rotated, and in each case after cleaning, dry the part thoroughly, and coat it with a light lubricating oil. The most suitable cleaning method is to soften the carbon first, by the following method, and then remove the final remains of the carbon with a rag or soft brush.

Immerse the article in a mixture of 30 percent of Oakite No. 9 and 70 percent of Diesel fuel. Soak for 12 to 24 hours while agitating the fluid with a stream of low-pressure air. Rinse off and then blow down with a stream of steam and water. Clean off any remaining carbon with a rag or soft brush.

INDICATOR CARDS AND COMBUSTION

Some service engines have a reducing mechanism which will permit obtaining closed pressure-volume indicator cards of the working cylinders. These diagrams show the work performed in the cylinder, the sequence of events of the working cycle, and, in conjunction with observation of the exhaust, aid in judgment of the combustion. Good combustion is shown by a clear exhaust, a broad diagram during combustion and a low end pressure on the expansion line. If the indicator diagrams show irregularities, hit-and-miss methods to remove the troubles should be strictly avoided. Make a systematic examination of the fuel system, valves, and timing. The proper values of firing and compression pressures should be obtained for each engine from the manufacturers instruction book.

MAXIMUM CYLINDER PRESSURE INDICATORS

Many of the higher speed engines are not fitted with indicator gear. In those cases, the indicators furnished may be used only to obtain the maximum firing pressure or cylinder compression pressure with the cylinder not firing. Complete directions will be supplied with each instrument. The manufacturer's instruction book will list the allowable range for compression and firing pressures. Low compression pressure indicates leaking valves, blowby from sticking rings, or excessive wear. Unusually high or low firing pressures indicate derangement in the fuel system.

MINOR REPAIRS MUST NOT ACCUMULATE

Do not permit minor repairs to accumulate. Every repair should be made as soon as personnel and materials are available. Unexpected operation orders may otherwise require the work to be hurriedly and carelessly done.

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2. REPAIR OF SPECIFIC ENGINE PARTS

ENGINE AIR COMPRESSORS

(1) Air compressor pistons should be withdrawn and cleaned every Navy overhaul. Third-stage rings should be removed for overhaul or renewal every 9 months. Examine the alignment of pistons and cylinders if the efficiency of the compressor falls off. See that the rings are not worn excessively, are free in the grooves, and exert the proper pressure against the cylinder walls. Give careful attention to air compressor valves. They must be removed, inspected and cleaned at least once every 150 service hours. Weakened springs and defective valves must be replaced. Excessive fouling of valves is due to too much lubricating oil, oil wiper rings not functioning properly, improper cooling or cleaning of the air and/or improperly drained crankcase.

(2) Improper operation of air compressors, especially valves, is recognized at once by a diminished pressure in the intermediate stages. The normal pressures are given in the instructions for each installation and should be practically independent of the speed of the engine and of the actual pressure in the spray air flasks.

(3) An abnormally high pressure on one stage indicates either that the discharge valve of that stage or the suction valve of the next higher stage is leaking or that the piston rings above this stage leak. The suction line becomes hot when the suction valves leak. If at any time the pressure or temperature in any stage is abnormal, the air compressor should be secured and not operated until the trouble has been located and remedied.

AIR COMPRESSOR COOLERS

(1) Air-compressor air coolers must be disassembled and the coolers and separators inspected and cleaned after 1,500 hours operation, or at any time when the coolers do not function properly.

(2) Small leaks around the tubes interfere with proper circulation of the cooling water. They may be detected by hot spots on the tops of the coolers. Large leaks will burst safety washers or open relief valves.

FUEL INJECTION VALVES AIR INJECTION ENGINES

(1) Before inspecting air-fuel injection valves, it is essential that the spray air lines be vented. The packing and guides should be well lubricated during routine examinations. Necessary precautions should be taken to insure that fuel is not pumped into the cylinder by the hand pump while removing the valve needles. The valve needles

must work freely and should fall on their seats by their own weight when rotated and released. When installing the valve needle, turn it back and forth; if it sticks in one place and moves easily in another, the spindle is bent and must be straightened or replaced.

(2) A dirty atomizer will give poor combustion and cloudy exhaust. The atomizer holes should be cleaned periodically with brass wire of the proper size.

(3) Fuel injection valve packing must not be set up while the engine is turning over. New packing must not be installed on top of the old packing when the stuffing box leaks. The old packing must be removed and the stuffing box cleaned.

(4) Unnecessary grinding-in of fuel injection valves is to be avoided. It is necessary only when the seat has been damaged and leaks. Use only the finest grinding compound when grinding-in fuel valves. Open the indicator cocks and admit spray air up to the valves to test for leaks.

FUEL INJECTORS AND PUMPS SOLID INJECTION ENGINES

(1) Fuel injectors and fuel pumps should not be torn down for overhaul unless defective operation on engine or test stand indicates this to be necessary, and then only those parts should be renewed which are found defective or worn beyond further economical use. Units should be opened up only to the extent necessary to effect repairs.

(2) Springs should be renewed only if visibly defective or permannently set beyond the limits allowed.

(3) **Never use grinding compound on the valves of a fuel injector.** Rotating the needles with filtered Diesel fuel is usually sufficient to free it and clean the seat. The liquid from metal polish, poured off after all but the suspended material has settled out, will contain sufficient abrasive to reseat badly carbonized valves. Clean the valve with filtered fuel after it is free.

(4) Clean the injector openings with a wire one to two thousands smaller than the diameter of the orifice. Do not use a probe which will break off and plug the hole.

(5) Keep all fuel drains and return lines open and clean. Renew all defective gaskets and those where effective sealing is doubted. Prevent any fuel oil leakage which might dilute the lubricating oil.

PISTONS AND PISTON RINGS

(1) Piston heads must be kept free from excessive carbon. The carbon in the combustion space may be burned off by a period of operation at full load. The carbon on the cooling surfaces on the underside of the piston must be cleaned manually. This carbon may

flake off and clog the lubricating oil filters and lines. It is advisable to remove this carbon whenever a piston is removed for inspection.

(2) The two uppermost compression rings should be renewed whenever a piston is removed from an engine. If otherwise satisfactory, the remaining rings shall not be renewed unless the gap clearance has increased 100 percent over the original installed clearance.

(3) The ring grooves should be cleaned without removing the rings whenever possible, as rings are distorted in removal and require many hours running-in before seating properly. The gap clearance should be measured with the rings on the piston, by compressing the rings to bore diameter in a sleeve having a port through which the gap may be measured.

(4) Rings should be installed by means of thin metal strips over which the rings will slide, or with the aid of an approved ring expander so that the rings will not be unduly stretched in assembly.

(5) Measure and record the ring side and butt clearances at each inspection. Generally a ring should be replaced when either its side or butt clearance becomes twice the maximum allowable installed clearance. A butt clearance below the minimum specified clearance may result in a piston seizure if the ring sticks in its groove and buckles when expansion closes up the gap. Do not use pistons in which the grooves are battered to such an extent that there is not a continuous bearing surface for a ring.

(6) In general, the minimum piston ring butt clearance should be 0.006 inch per inch of cylinder diameter. Many engine manufacturers supply rings with a smaller gap than this to obtain longer ring life. If piston ring trouble is experienced and it is suspected that the ring ends are butting, the end clearance should be increased to the above minimum.

(7) Examine piston rings through the ports on two-cycle engines if practicable. The faces of stuck or broken rings will be black and the piston must be pulled and rings renewed at the first opportunity.

PISTON PIN NEEDLE BEARINGS

Needle bearings on piston pin assemblies should be rejected if they have any surface defects. One failed needle, possibly initiated by such defects, may enter the lubricating oil and cause failure of several bearings or a piston seizure. If some rollers must be replaced because of surface defects, the entire set must be replaced unless they are worn less than 0.003 inch; otherwise the new rollers will be overstressed.

BEARINGS

(1) Precision and babbitt bearings should be renewed only when wiped, worn beyond limits allowed, honeycombed to such an extent that complete failure is imminent, or when sufficient bearing metal is broken loose from the shell to result in a serious reduction of bearing area. Circumferential scratching and grooving, often noted during inspections, is caused by improper fitting at the bearing joints, allowing particles of the bearing metal to be rolled off, or by dirt in the lubricating oil or in all or any part of the lubricating oil system.

(2) Precision main bearings should be renewed as a set to make sure that the new bearing will not be a high spot in the alignment of the crankshaft and hence take undue wear. Precision bearings should not be scraped.

(3) Babbitt bearings should be spotted on the crankshaft with the maximum thickness of shims installed and then scraped to a uniform bearing. An equal shim thickness should be used on each parting face of the shell. The bearing clearance should then be checked with leads and, if required, increased by further scraping or decreased by removing shims.

UNDERSIZED BEARINGS

(1) When necessary to undercut crankpins to clean up a scored journal, they should be undercut in increments of one-thirty-second inch (0.03125 inch). The Bureau of Ships has available special portable tools for refinishing crankpins in place, for use on the following engines:

General Motors Models 201A, 248, 278, and 567.
Hooven, Owens, Rentschler Model D–Z.
Busch-Sulzer Model DHBM.

(2) In some instances, it is feasible to repair damaged crankshafts by the metal spray process. The approval of the Bureau of Ships should be obtained before such extensive repairs are made.

GEARS

Gears should not be renewed merely because of initial pitting of teeth. If the bearing areas on the teeth show that alignment is correct, the gears can be used until wear and backlash become excessive for the service required, or the pitting extends to the point of possible tooth failure. In certain cases, gear tooth pitting does not continue after the bearing surfaces are well worn in, following the first few hundred hours of operation, and such gears may be retained in service, provided all operating requirements are met and the gear is inspected for condition at regular intervals.

GOVERNOR

The governor should be washed every 400 hours with clean fuel oil. Drain, flush and refill with clean lubricating oil. Carefully inspect all parts of the governor and its connecting linkage, for excessive play or for binding. Likewise, inspect the parts of the engines fuel mechanism to which the governor is connected. In order to do this, remove the pin at the governor, and try the governor linkage and the engines fuel mechanism separately. Excessive play is eliminated by tightening loose fasteners or by replacing worn parts. If binding is noted in the governor remove the pins in the linkage, one at a time, until the point of binding is located. A similar procedure will serve to locate sources of lost motion or binding in the engine's fuel mechanism. If it is necessary to disassemble the governor, first secure a copy of the engine manufacture's instruction book and follow the instructions outlined therein. When reassembling the governor, use only hard grease on the gaskets. Under no circumstances should shellac be used.

BALL OR ROLLER BEARINGS

Ball or roller bearings should be renewed only when the bearings or races are definitely known to be pitted or badly worn. Frequently, fresh lubrication is all that is required after a thorough cleaning has removed gummed lubricating oil or grease.

LEAKAGE OF CYLINDER HEADS

Cylinder heads should be renewed when they are obviously cracked, or when a pressure test shows them to be so. Repair by welding should not be attempted. Cracks are manifested by leakage, externally, or into the valve chamber or combustion space.

LEAKAGE OF INLET AND EXHAUST VALVES

Air intake and exhaust valves should be carefully examined for cracks, fouling, tightness, and seat condition, and ground in, if necessary, whenever they become available because of some other inspection. It is not possible to give a definite period between inspections because of varying conditions for different types of engines, but the valves should be examined whenever there is the slightest indication of loss of compression. It is probable the maximum interval should be 500 hours. Grind in valves only until a gastight seat is obtained. Continued grinding to remove small pits requires additional work, results in a needless reduction in life of valves and seats, and is not necessary to seal the gases.

INLET AIR PASSAGES AND SCAVENGE RECEIVERS

(3) Air-intake ducts to working cylinders, scavenging pump cylinders, and air compressor cylinders should be kept clean so that air passage is not restricted and engine efficiency thereby reduced. Oil must not be allowed to accumulate in the scavenging receivers, as explosions are likely to occur.

CYLINDER WEAR

Maximum cylinder wear limitations cannot, with economy, be set for all engines in general due to variations in operating conditions which permit one engine to operate satisfactorily with wear which in another would result in blowby or ring breakage. Cylinders normally will not require renewal unless worn in excess of the following:

(*a*) Two-cycle engines with aluminum pistons, 0.0025 inch per inch of diameter.

(*b*) Slow-speed engines over 18-inch bore, 0.005 inch per inch of diameter.

(*c*) All other engines, 0.0035 inch per inch of diameter.

ALIGNMENT

A crankshaft failure or excessive main bearing wear will result from crankshaft misalignment. Bridge gage readings and crankweb deflections should be checked during major overhaul with particular attention given to deflections at the after main bearing.

THERMOCOUPLES

Thermocouples stems become coated causing the pyrometer to read low and lose its quick response to sudden changes to exhaust temperature. The stems should be cleaned periodically and checked by immersion in a liquid of known temperature.

EQUALIZING THE LOAD BETWEEN CYLINDERS

(1) Specific instructions for equalizing the load between cylinders are issued for each engine. In general, the following practice should be used for engines with General Motors unit injectors, Bosch fuel pumps or similar pump injector systems:

(*a*) Before attempting to equalize the cylinders, be sure that all the injectors and fuel pumps are properly assembled and that the fuel pump plungers are correctly timed with the engine in accordance with the engine builder's instruction book.

(*b*) Then, block the governor in the wide open position and with injector control shafts in maximum fuel position, set the racks exactly equal at the full load setting.

(*c*) Check governor linkage to injector control shafts. Be sure that with throttle in off position the injector racks are in "no fuel" position and that the travel of the governor power piston from off to full load position is correct. The injectors are now equalized so that they will deliver equal fuel quantities for all rack positions within manufacturing tolerances.

(*d*) Start the engine and bring it up to cruising load. After steady running conditions are attained, equalize the exhaust temperatures by minor adjustment of rack settings. This adjustment should only be made after the pyrometer equipment has been checked and thermocouples and master switch contacts have been cleaned.

(2) When the injector pumps have been adjusted for cruising load, rack settings are not to be changed merely to equalize exhaust temperatures. Some variations in exhaust temperatures are normal and to be expected when changing load. Pyrometer temperatures should only be used to indicate trouble and the condition of the engine. Wide variations in exhaust temperature may be caused by one of the following:

(*a*) Faulty fuel injector or fuel pump.
(*b*) Plugged injector fuel filter.
(*c*) Carboned or defective thermocouple.
(*d*) Defective pyrometer or thermocouple lead.
(*e*) Burned or sticky valves.
(*f*) Excessive ring blow-by.

(3) A variation in exhaust temperature of 50° is normally permissible; however, the maximum allowable exhaust temperature given in the manufacturer's instruction book should not be exceeded.

TYPICAL TROUBLE SYMPTOMS

Operating personnel must rapidly become accustomed to all sounds and noises emanating from their engines. Each sound has it's own significance, and any variation should be noted and investigated as a possible warning of trouble. Some of the more pronounced "sound symptoms" are:

(a) A pounding sound which occurs each stroke (twice engine speed) and grows progressively louder may be caused by a *piston approaching seizure.* The engine speed should be cut in half immediately by reducing the load; if the condition disappears the engine possibly can be operated at reduced load for a short period. If the trouble does not disappear the engine must be stopped, slowly cooled, and the piston pulled for repair. The binding is probably caused by poor heat transfer from the piston through stuck piston rings.

(b) An alternately loud and soft pound may be due to *loose connecting rod bolts* or a *loose crankbox*. Exact source of trouble must be located and fixed immediately.

(c) A slapping sound which occurs every revolution on a two-stroke cycle engine, or every other revolution on a four-stroke cycle engine, indicates a *broken valve spring*. It should be repaired as soon as possible, although limited operation *perhaps* can be continued in an emergency.

(d) A dull thud during combustion stroke usually indicates either *late injection timing* or a *faulty spray nozzle*.

(e) A sharp knock during starting up may be a fuel knock, due to low engine temperature. After an engine has reached proper operating temperature, persistence of such a knock may indicate too-early injection timing.

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